Heating·Piping
and Air Conditioning

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Diesel Plant and Laboratory Salvage Electricity, Use It for Space Heating

By A. F. Davis*

What would otherwise be wasted current from diesel engines on test is salvaged at General Motors' new diesel manufacturing plant and laboratory by a unique heat and power pool. . . . With the arrangement—described here—electric current produced by the engines on test is used to generate steam for plant heating and process uses; the laboratory is heated and ventilated by a blower system distributing air which is heated by passing over electrical grids

IT has been said that modern meat packing methods are so efficient there is nothing wasted but the squeal of the pig. Those in the packing industry may decry the statement as an exaggeration, but it certainly is apropos of General Motors' new Detroit Diesel Engine Division plant and the adjacent diesel laboratory, if the scheme adopted for the utilization of what otherwise would be wasted electrical current can be taken as a criterion.

The 240 x 480 ft manufacturing plant (Fig. 1) is devoted to the production of the small series of General Motors two cycle diesel engines, and the 105 x 223 ft laboratory (Fig. 2) to the development of these and all other GM diesel units, including the larger sizes built by plants at Cleveland, Ohio, and LaGrange, Ill. Steam is required by the plant for heating purposes, as well as for certain manufacturing operations. It is used, for example, to keep the tin and copper plating solutions at the required temperatures, and for the quenching tanks in the heat treating room. It is employed likewise at cleaning stations along the production lines, and for the purpose of maintaining the proper water temperature in engines being run on production test.

Excess Current Generates Steam

All engines in the laboratory, as well as those on the 10 test stands in the manufacturing plant, are tested by electrical brakes, either "run in" generators or cradle type dynamometers. Ordinarily, of course, current produced by such engines is wasted, being dissipated in the form of heat by its passage through resistance grids. Excess electrical current developed by the new diesel laboratory and plant, however, is used to generate steam, being salvaged into a single system with this in view. In addition to eight dynamometer rooms, the laboratory contains diesel powered generator equipment and two steam boilers, one oil fired and one electric, the later containing 186 heating elements capable of dissipating nearly 2800 engine horsepower. (Fig.3).

Both boilers are connected to a single steam feeder. They are fed by the same boiler accessory equipment, feedwater heater, feedwater pumps, etc. The oil fired boiler (Fig. 4) is controlled automatically after starting, and is used only when the electric boiler cannot supply sufficient steam.

Fig. 5 shows the electrical hook-up. Three different voltages must be handled. Alternating current is generated at 4600 volts, while standard railroad generators are designed for 550 volts d-c and electric dynamometers are constructed for 275 volt d-c operation.

The laboratory and production test stands generate direct current, as do the marine and railway engines. Stationary engines, on the other hand, usually are required to generate alternating current, and must be tested for the production of that type of power.

To insure complete flexibility, the electric boiler is divided into three sections, two of 550 volt capacity and one of 275. Power generated by the 8 and 12 cylinder engines now installed in the power room (Fig. 6) may be sent through the motor generator set, or direct to the electric boiler, in which case the boiler serves as an electric absorption dynamometer.

The 16 cylinder engine, which operates at 720 rpm, is connected to a 1000 kw alternator. This is a typical stationary installation. It is so arranged that its output may be dissipated in the electric boiler for test purposes by first being passed through the motor generator set. It will be seen that the big, three unit motor generator set, which is of 3500 kw capacity, forms the heart of the system.

When the production and laboratory test stands are "pumping back," the power engines are first relieved of their

Fig. 1 (left)—The plant. Fig. 2 (right)—The laboratory

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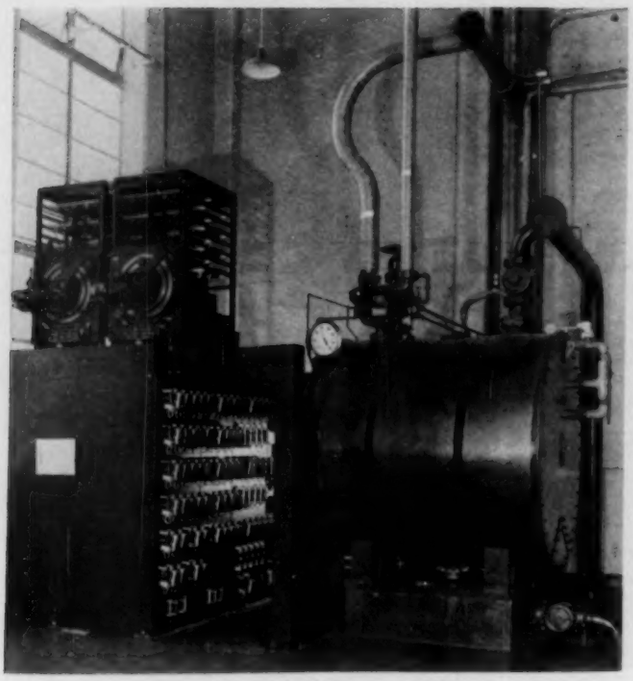


Fig. 3—Electric boiler is capable of dissipating nearly 2800 engine horsepower

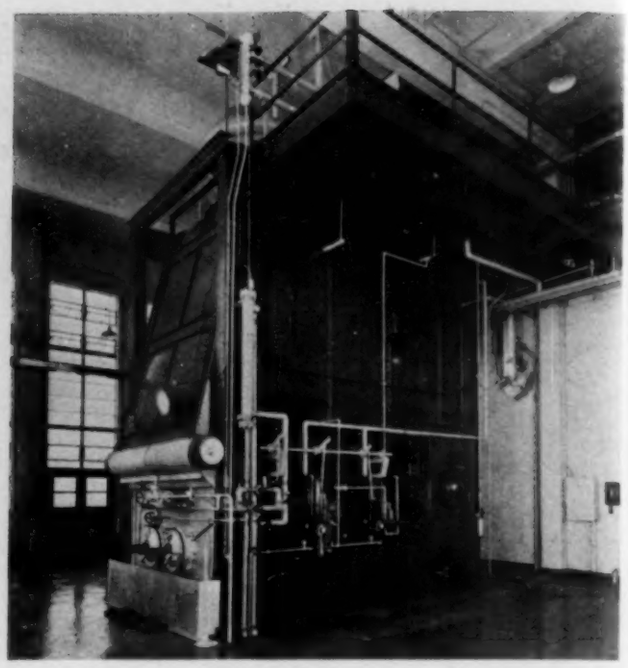


Fig. 4—The oil fired boiler, which is used when the electric boiler cannot supply sufficient steam

load. If there is still an excess, a frequency relay automatically shunts it into the 275 volt section of the electric boiler. This relay is designed to act as a balance and keeps the system as a whole on an even keel.

Heating the Laboratory

The method of heating and ventilating the laboratory is of particular interest, for there are no radiators or steam pipes, save those leading to the offices at the extreme end of the structure and to the manufacturing plant.

Radiant heat from engines undergoing test and that from normal losses in the electrical system is distributed throughout the laboratory by means of blowers. A small grid room employed in conjunction with certain tests, where there is a continual fluctuation in current and voltage, is tied in with this system. Efficiency, of course,

Fig. 5—Diagram of the electrical hook-up

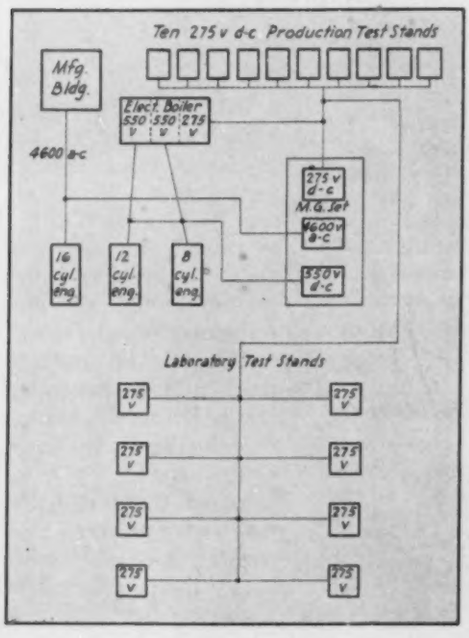


Fig. 6—Diesels in the power room are tied in with the pool

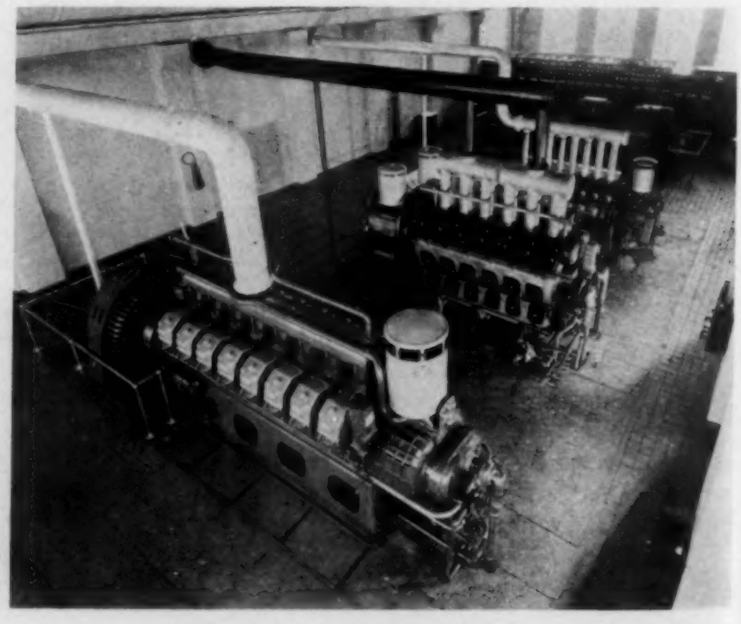
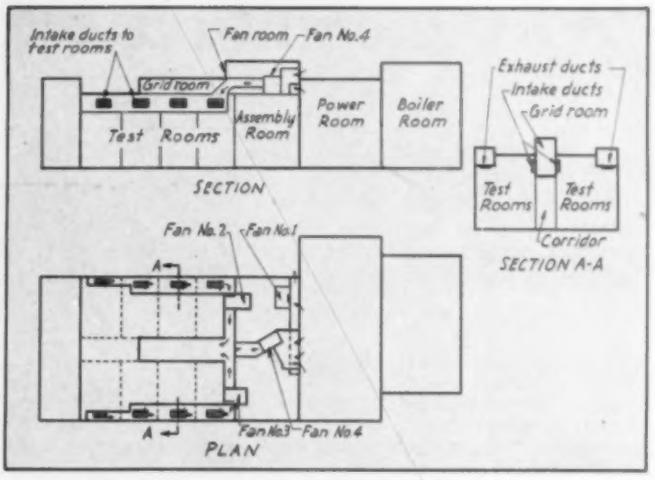


Fig. 7—Diagram of system for heating and ventilating the laboratory



is increased by the fact that there are no doors or windows leading outside the building from the power or test rooms.

Fig. 7 diagrams the system. Fan No. 4 sucks warm air out of the power room, forcing it down a duct formed by the false ceiling in the central corridor and into the dynamometer rooms. It is drawn out through exhaust ducts at either side of the structure by fans Nos. 2 and 3.

Temperature is controlled by bypassing through the grid room. This makes it possible to temper outside air brought into the building, to raise the temperature of the air drawn in from the power room, or to recirculate the air within the structure as desired.

In the summer months, fan No. 1 is brought into play. This draws air out of the power room, discharging it into the atmosphere. At the same time, fan No. 4 brings air in from outside, forcing it into the test rooms. Fans Nos. 2 and 3 perform the same function as at other seasons of the year, but in this instance exhaust the air through the grid room as a means of cooling the grids.

City Air Searched for Sulphur Fumes

RESULTS of a 15 month survey to determine the *average* amounts of sulphur gases in the air of American cities are announced by Air Hygiene Foundation and Mellon Institute. The average amounts of sulphur fumes found in 25 cities studied were comparatively small.

50,000 Separate Tests

More than 50,000 separate air tests were made by six chemists driving a fleet of cars equipped like "traveling laboratories." They canvassed industrial and residential centers throughout the east, south and midwest, from August, 1936, to October, 1937. Most of the tests, covering all hours of day and night and all seasons of the year, were made in five metropolitan districts. The five, in the order of their sulphur dioxide pollution are (figures indicate parts of sulphur dioxide per million parts of air within 15 mile radius of center of city):

	Average	Maximum
St. Louis-East St. Louis	0.128	2.266
Pittsburgh	0.057	0.897
Detroit	0.028	0.396
Philadelphia-Camden	0.027	0.424
Washington	0.009	0.290

The above figures are of no significance from a public health standpoint, according to hygienists connected with the investigation. That is, "in the concentrations found, the contaminants do not exert harmful physiological effects."

The survey showed that the home fires are among the large contributors to sulphur pollution, particularly in districts using coal of high sulphur content. The type of coal burned in a locality was mirrored in the results of the study. The fuel factor also explains why sulphur pollution in most districts was approximately 50 per cent higher in the heating season than in the summer months. Some industrial operations also discharge sulphur fumes unless properly safeguarded.

Wind Cleans Air

A close relationship was found between wind velocity and the amount of sulphur dioxide in the air. The higher the wind the cleaner the air. Fogs catch and "store up" the sulphur fumes. Some of the highest concentrations were noted on foggy nights.

Occasional tests were made in a score of other cities. Dr. H. B. Meller, managing director of the Foundation, cautioned that results obtained in these cities *cannot be compared* with the findings for St. Louis, Pittsburgh, Detroit, Philadelphia-Camden, and Washington. He pointed out that "only a few tests were made in this group, not enough to arrive at a typical, average figure, as in the case of the five centers which formed the backbone of the survey." Results for the 20 other cities:

	Summer Average	Winter Average
Baltimore	0.021	0.081
Chicago	0.067	0.091
Cleveland	0.064	0.081
Wheeling	0.070	
Nashville	0.028	0.093
Cincinnati	0.021	0.064
Buffalo		0.044
Youngstown	0.049	0.026
Louisville	0.022	0.041
Ft. Wayne	0.028	
Richmond	0.009	0.047
Indianapolis	0.023	
Toledo	0.023	
Chattanooga	0.011	0.035
Springfield, Ill.	0.021	
Birmingham	0.017	0.017
Charlotte, N. C.	0.015	
Johnstown	0.014	
Harrisburg	0.011	
Atlanta	0.012	0.032

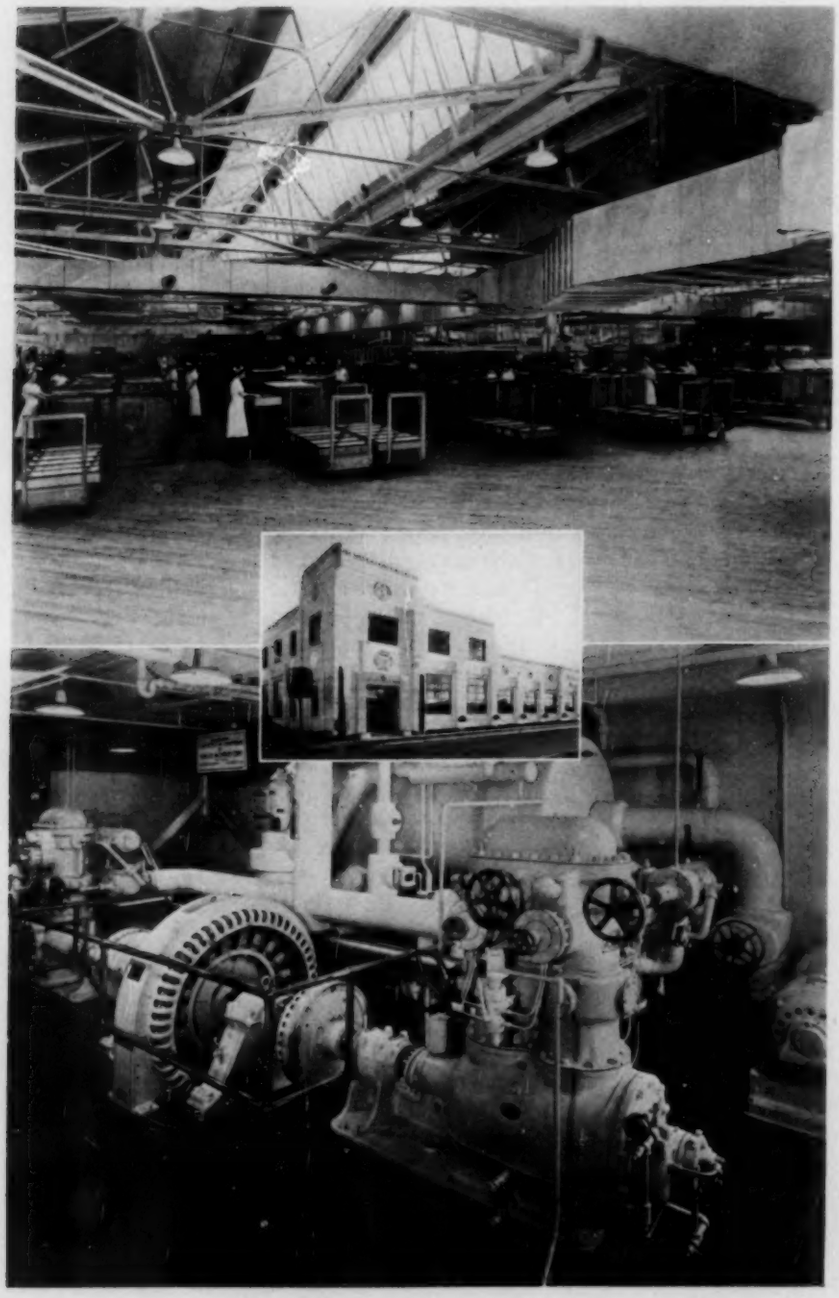
Why Condition a Bakery?

By W. E. Barnum*

The Helms Olympic Bakeries—said to be the first completely air conditioned bakeries — answer the question by their experience. A uniform product requires uniform control of all ingredients, which in turn requires control of the air conditions . . . . . There are also several incidental benefits

WHY air condition a bakery? The question is answered by the famous Helms Olympic Bakeries, Los Angeles, Calif., the first completely air conditioned bakeries in the world. A uniform product has far more appeal than one which fluctuates daily. Helms Bakeries realized that uniform control of all ingredients cannot be obtained without regulating air conditions in all processes. In the past, bakers have confined their air conditioning efforts to the dough and fermentation rooms, the proof boxes, and bread coolers.

Inset—The completely air conditioned Helms Olympic Bakeries . . . . . *Upper view*—Main and branch ducts in the cake division, showing the nozzle type high velocity outlets. Air is regarded the same as any other ingredient that goes into the manufacture of the products . . . . . *Lower view*—Two compressors operated by a 250 hp synchronous motor comprise the condensing system. The motor can drive one or both compressors or run free to improve plant power factor, by means of a clutch



System Split Into Five Zones

The Helms installation is a 275 ton central station, water cooling air conditioning system, with a condensing system having two compressors operated by a 250 hp synchronous motor—a flexible arrangement which will operate at 100, 75, 50, and 25 per cent capacity to permit different parts of the system to operate independently. The rest of the system consists of water cooler, dehumidifier, chilled water pump, sump pump, two condenser water pumps, forced draft spray cooling tower; and booster fans, heaters, ducts, controls, etc., for each individual zone.

The air conditioning system is split up into five zones as follows: 1. Private offices. 2. General offices. 3. Mezzanine floor of bakery. 4. Main floor of bakery. 5. Bread coolers. The central dehumidifier supplies conditioned air to the first four zones and chilled water lines from the central water cooling system carry refrigeration to the bread coolers.

The private and general offices are year 'round conditioned to maintain 75 F and 50 to 55 per cent relative humidity. The private offices are on a separate zone because they have an outside exposure and consequently conditions vary more widely there. In all offices conditioned air is introduced through ceiling outlets incorporated in the overhead lighting system.

The bakery proper consists of the receiving room, flour storage, pastry department, mixing room, bread department, cake department, oven space, bread loading and cake load-

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ing spaces—a total area of 97,500 sq ft where proper temperature and humidity control is most essential to uniform production. This area is maintained at 80 F and 50 per cent relative humidity and is handled as one zone with a complete air distributing system arranged for year 'round air conditioning. The main heat load is due to the traveling ovens and is so large that exhaust fans carry most of it directly to the outside. In winter these fans are shut off and the oven heat is recirculated to provide heating for the whole plant.

Bread Cooling Important

In applying refrigeration to the bread cooling system, the air conditioning engineers were surprised to learn that there was little available information to work with. Bread cooling has been a hit-or-miss proposition with bread being stored on racks and subject to any room temperatures from 75 to 100 F and any prevailing humidity or mold spores in the air. In addition, variable air currents throughout the racks made uniform cooling impossible. Helms Bakeries found that the bread wrapped before proper cooling is apt to mold and also that high humidity during cooling has a decided effect on the softness of the bread. In an attempt to control cooling in the bread cooler, Helms had previously installed an air washer and filters for air supplied to this cooler. In spite of this equipment, variable outside conditions would often raise the humidity of the washed air to a point where undesirably soft bread resulted. After a complete survey of these conditions, the engineers revamped the washer to incorporate refrigeration by means of chilled water, and in addition installed a new cooler of slightly different design than the old one. The system is now able to maintain the required constant temperature and humidity at all times regardless of outside conditions. Better cooling is accomplished without using so much outside air.

Cooling is a very important part of the baking process; a Chicago bakery which installed air conditioning reports that their cooling time has been cut almost 15 per cent and it is their firm belief that this faster cooling results in better pies. The inside steam, they believe, has less chance to continue cooking the crust and as a result the pies are dry and crisp rather than damp and soggy. The effect is a decided improvement in the appearance and quality of the pastry, resulting in an increase in sales for this establishment.

The Helms bread wrapping mezzanine is air conditioned for year 'round control of temperature and humidity. The same conditions must be maintained here as in the bread cooler to prevent any change in temperature or humidity which might affect the bread inside the wrapping. Air is supplied by overhead ducts on the side walls, blowing toward the center.

In summing up this air conditioning system there are several noteworthy points. First, the method of zoning allows individual operation and flexibility so that the system can take care of variation in requirements at different peak conditions. Second, the method of using oven heat for winter heating reduces the size of the winter equipment considerably. Third, the synchronous compressor motor, of leading power factor, helps to balance the lagging power factor due to the operation of a number of small induction motors in the bakery, hence improving the power factor for the entire plant. In fact, this synchronous motor may be run without either compressor in operation, just for this purpose; this is possible because electric clutch connections from motor to compressors permit the motor to operate one or both compressors, or run free if necessary.

Is Complete Bakery Air Conditioning Justified?

Has this installation justified itself? The best answer is to quote Pres. Paul H. Helms, who says: "In the first place, we have absolute control of humidity in our shop, which is a great help indeed because the humidity in any bake shop varies considerably—not only with the outside temperature and daily air conditions, but also with the volume of merchandise manufactured. For instance, recently we had to bake a large order of hot cross buns hours ahead of time and hold them in the shop until we were ready for delivery. At this particular time we raised the humidity in the shop to keep the buns moist and to prevent drying out. Most of the time, however, we reduce our humidity so that the merchandise does not sweat and so that the bread and rolls keep a crisp crust, and we find that having this humidity control not only in our fermentation room and proof case, but also in the shop itself, including the bread cooler, is a great boon to our manufacturing problem.

"One of the greatest benefits we have is the absolute uniform cooling of our bread regardless of outside conditions, which can only be accomplished with refrigeration. Depending on the cooling from the water sprays alone is not enough. This past summer we had scarcely one complaint of bread mold, and of course some days were very hot and humid.

"As I look back upon it now, I would make this entire investment for the better employee relationship. We have the happiest baking crew I have ever seen in any one shop in my 25 years of baking experience. In addition the efficiency of our people has increased about 10 per cent, as closely as we can figure.

"Air conditioning has gone a long way in insuring cleanliness in our plant—dust is taken out of the air and the air is cleansed and recirculated. We find it much easier to keep our shop clean. Dust is not even found in our flour sifting room.

"The air in our bakery—because of air conditioning—is handled just like any other ingredient that goes into the manufacturing of our products. We have a man that makes an hourly trip checking conditions in all parts of the shop, marking down in all departments the humidity and temperature. Our wrapping department, where we have five electrically heated wrapping machines, is ideal. We have no heat—we have no trouble sealing paper—we practically never have re-wraps—we save time and paper.

"One of the greatest advertising features we have is the air conditioning in our bakery."

Recent advances in oil burner design may make possible the renaissance of the steam automobile, which was in a poor competitive position in the early automobile market, partly because of difficulties with its firing equipment, according to Arthur D. Little, Inc.'s "Industrial Bulletin." Now, greatly improved gun type oil burners and more automatic controls make these cars again feasible.

Swivel Joints Simplify Process Piping

By Oliver J. Haller*

A UNIQUE arrangement of piping and swivel joints was recently fabricated for the purpose of transferring fluid from one vessel to another. The illustration shows the installation in a sulphate pulp mill where each digester serves six diffusers or washers. The specific problem here was to so arrange the piping and connections that the discharge from the digester could pass through any two of the diffusers in series.

Since the digester is blown with steam at a pressure ranging from 125 to 150 lb per sq in., and the diffusers operate at atmospheric pressure during the pulp washing process, it is necessary to flash off the excess heat of the blow. The flash vapor is therefore permitted to pass into a second diffuser, which also acts as a save-all for any pulp that is carried over. From this second diffuser the flash vapor passes to a condenser where the remaining condensable vapor is recovered and the non-condensable vapor passes to the atmosphere.

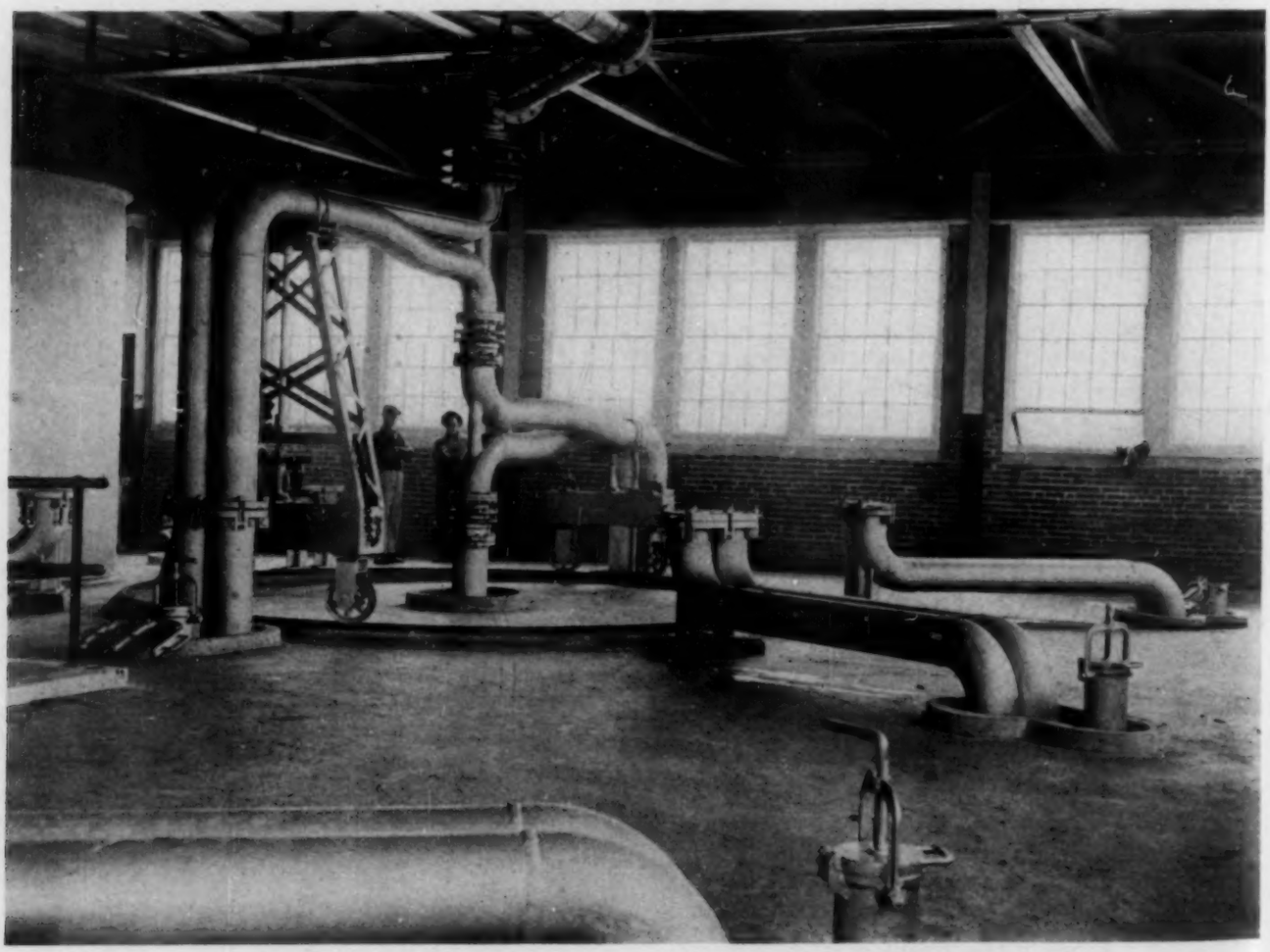
In this particular installation the diffusers are placed in two rows of three tanks each. This arrangement requires permanent connections from the four most distant diffusers to positions that are the same distance radially from the center of the digester as are the two middle tanks. A simpler installation could be effected if the tanks were located in a concentric circle around the digester.

The discharge pipe of the digester, the pipe leading from the first diffuser to the second, and the pipe entering the condenser, are each fitted with a swivel joint to permit connection of the digester to any two of the six diffusers. The pipes are rotated to the desired positions by two independent motor driven carriages operating on a circular track. Each carriage is equipped with a lifting jack for raising the pipe so the joint can be disconnected.

Except at the swivel joints, and connections to the fixed pipes leading from the diffusers, the entire piping system is of welded construction to reduce weight to a minimum. The welding elbows are steel castings with reinforced back so they can resist the erosive effect of particles of pulp at the bends. The glands and stuffing boxes are lined with stainless steel to prevent erosion.

*Engineer, Pittsburgh Piping and Equipment Co.

Unique arrangement of piping and swivel joints enabling discharge from digester to pass through any two of six diffusers in series. Pipes are rotated to the desired positions by motor driven carriages running on the circular track



Figuring Solar Heat Gains of Buildings

By William Goodman*

IN the May and June issues, tables were presented—and their use explained—for computing the solar heat gains through walls, roofs, and windows. The theory underlying the subject of solar heat gains and the methods used in preparing the tables were discussed in July and August and are continued this month.

Theory of Solar Heat Gains Through Walls and Roofs

The solar heat gain through walls and roofs cannot easily be computed with any degree of accuracy. Although methods are available for accurately estimating the heat gains through walls and roofs, they are not suitable for the everyday work of estimating. Furthermore, even if one were inclined to use these methods, the values of I_p, and the physical properties of many building materials—such as density and specific heat—are not known with a sufficient degree of precision to warrant the use of elaborate computation procedures.†

The amount of radiant energy that will probably fall on the outside surface of a wall or roof can be computed by means of Equation 3. But of this amount only a relatively small percentage reaches the interior surface of the wall, there to be transferred to the room air. First, a certain amount of radiant energy is reflected back into space from the outside surface of the wall. The amount reflected and the amount absorbed by the surface depend upon its color and texture. The heat absorbed by the surface raises its temperature to a point higher than the temperature of the surrounding outdoor air. Because of this, some of the solar heat absorbed by the surface is transferred to the outdoor air in contact with it. Of the remaining portion of the solar heat that starts the trip into the interior of the wall, only a fraction ever reaches the interior surface. The reason for this is to be found in the heat capacity of the structure itself. When sunshine first falls on a surface, it raises the temperature of that surface. Heat is conducted from the outside surface layer of the wall to the layer underneath. Therefore, the temperature of this layer increases also, and it in turn transfers a fraction of the heat that it receives to the layer underneath it. Each layer transfers to the next one slightly less heat than it receives because some of the heat is retained in raising the temperature of the layer itself. Eventually the temperature of the inside surface of the wall is raised sufficiently so that heat is transferred from this surface to the air in the room.

Because the temperature of the entire mass of the wall must first be raised before the inside surface temperature of the wall can be increased by solar heat, the maximum flow of heat from the interior surface of the wall to the air in the room takes place some hours after the solar energy falls on the outside surface at the maximum rate. As previously stated, the number of hours required for heat to travel through a wall is known as its *time lag*.

A wall acts as a reservoir for the storage of solar heat. When sunlight first falls on the surface of a wall, heat is stored within the wall as the temperatures of the various interior layers of the wall increase. Later when the rotation of the earth carries the surface out of the sunlight and into the shade, the temperature of the wall falls as it loses heat to the cooler outdoor air. Because of the reservoir-like action of the wall, only a fraction of the heat that starts the trip into the interior of the wall ever reaches the room. From an analysis presented in Appendix 9 of some A.S.H.V.E. tests, it is probable that not over 50 per cent of the solar heat that starts the trip into the interior of a wall or roof of ordinary construction ever reaches the inside surface of the wall This value is offered not as a definite and accurate one, but only as one that may be regarded as safe for estimating the probable maximum solar heat gain through walls and roofs of ordinary construction. This method of assuming a constant percentage for the solar heat transmitted through a wall cannot be defended on theoretical grounds, but in the present state of our knowledge, it seems to offer the only simple workable method of attacking the problem.

By analyzing the flow of solar heat through a wall, and regarding the percentage transmitted as constant, the following formula is derived in Appendix 9 for computing the solar temperature differences of walls and roofs.

$$D_s = a \frac{P}{f_o} I_u - (1 - P)\ D \qquad [13]$$

In computing Tables 1 to 8, the following constant values were used in Equation 13: $f_o = 2.0$, $P = 0.5$, $D = 15$ deg. For dark-colored surfaces, $a = 0.9$. For medium-colored surfaces, $a = 0.7$.

Substituting these values in [13], the following equations result. These equations were used in computing Tables 1 to 8.

For dark walls: $D_s = 0.225\ I_u - 7.5$ [14]

For medium colored walls: $D_s = 0.175\ I_u - 7.5$ [15]

By means of Equation 13 it is a simple matter to derive working equations similar to [14] and [15] for surfaces having different values of a, P, or D. For light colored, smooth surfaces, a value of $a = 0.4$ is customarily used. Values of a for a large number of surfaces of different colors and textures are listed by Brooks[6]. The uncertainty as to the condition of a surface over a long period of time because of the dirt and weathering, makes it difficult to select the value of a with any accuracy. For this reason, it is probable that the values of D_s

*The Trane Co. Member of Board of Consulting and Contributing Editors.

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† See list of symbols on p. 446, July issue.

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[6] Brooks, R. A., Solar Energy and Its Use for Heating Water in California, Bulletin 602, University of California, Table 3, p. 18.

listed in Tables 1 to 8, which were computed for a values of 0.9 and 0.7, will prove satisfactory in most instances for estimating the probable solar heat gain.

Where temperature differences other than 15 F are used in computing the conduction heat gain, the values in Tables 1 to 8 can be corrected by means of the following equation derived from [13]. Where the design temperature difference D does not differ much from 15 F, the correction may be ignored.

$$D_{s2} = D_{s1} + (1 - P)\ (D_1 - D_2)$$
$$D_{s2} = D_{s1} + 0.5\ (15 - D_2) \quad \text{.....[16]}$$

D_{s1} = tabulated solar temperature difference;
D_{s2} = corrected solar temperature difference;
D_1 = 15 degrees Fahrenheit;
D_2 = assumed design temperature difference.

Example 6 illustrates how the solar temperature differences in Tables 1 to 8 were computed.

Example 6: Find the solar temperature differences for a dark colored wall facing west in a latitude of 35 deg at 2 p. m.

Solution:

From Table 9, $\cos i = 0.4698$

From Table 10, $I_p = 302$ Btu

$$I_u = I_p \cos i \quad \text{.....[3]}$$
$$= 302 \times 0.4698$$
$$= 141.9$$
$$D_s = 0.225\ I_u - 7.5 \quad \text{.....[14]}$$
$$= 0.225 \times 141.9 - 7.5$$
$$= 24 \text{ F.}$$

Occasionally when using Equations 14 and 15 for computing the solar temperature differences in Tables 1 to 8, negative values were obtained. A negative value of D_s means that the total solar heat gain is less than the conduction heat gain when the surface is in the shade. Because of the present state of knowledge concerning the solar heat gain through walls and roofs, it seems best not to use any value of the total solar heat gain that is smaller than the conduction heat gain. For this reason, negative values of D_s are not given in the tables. In such cases, the minimum solar heat gain is assumed to have the same value as the conduction heat gain.

Appendix 1—Derivation of Equation 3

$$I_u = I_p \cos i$$

The total radiant energy in a beam of sunlight at the surface of the earth is the same at all sections of the beam. Referring to Fig. 1, the total solar energy passing through the imaginary perpendicular plane *3-4-5-6* is equal to the total energy falling on the plane *1-2-3-4*, or ,

$$I_u A = I_p A_p$$

A = area of plane *1-2-3-4* in Fig. 1.
A_p = area of plane *3-4-5-6* in Fig. 1.

$$I_u = I_p \frac{A_p}{A} \quad \text{.....[a]}$$

But $$\frac{A_p}{A} = \frac{3-6}{2-3} = \sin h \quad \text{.....[b]}$$

Also $$h = i - 90$$
$$\sin h = \cos i \quad \text{.....[c]}$$

Substituting [b] and [c] into [a],

$$I_u = I_p \cos i$$

Appendix 2—Derivation of Equation 5

$$\cos i = \cos h \sin (z - \theta)$$

Referring to Fig. 4,

$$\frac{1-5}{1-3} = \cos h \quad \text{.....[a]}$$

$$\frac{1-4}{1-5} = \cos a \quad \text{.....[b]}$$

Multiplying [a] and [b],

$$\frac{1-4}{1-3} = \cos h \cos a \quad \text{.....[c]}$$

But $$\frac{1-4}{1-3} = \cos i \quad \text{.....[d]}$$

Equate [c] and [d],

$$\cos i = \cos h \cos a \quad \text{.....[e]}$$

Referring to Fig. 3,

$$a + z = \theta + 90$$
$$a = \theta - z + 90$$
$$\cos a = \cos [(\theta - z) + 90]$$
$$= -\sin (\theta - z)$$
$$= \sin (z - \theta) \quad \text{.....[f]}$$

Substituting [f] into [e],

$$\cos i = \cos h \sin (z - \theta)$$

Appendix 3—Derivation of Equation 7

$$P_{T1} = (1 - P_r)^2\ e^{-kL}$$

When a ray of light is intercepted by the boundary surface between two different substances, part of the light passes the boundary and enters the second substance; the remaining portion of the light is reflected by the boundary surface. This is illustrated in Fig. 8 where a ray of light of energy I_o is intercepted by the outside surface *A-A* of the glass. Part of the ray passes the boundary *A-A* and enters the glass with a diminished energy I'_o and the balance of the ray of energy R_o is reflected from the surface.

In the same way when the light passes from the glass back to the air, it is intercepted by the boundary surface *B-B*. A part of the ray I_1 passes from the glass into the air, and the balance, of energy R_1, is reflected back from the inside surface *B-B*.

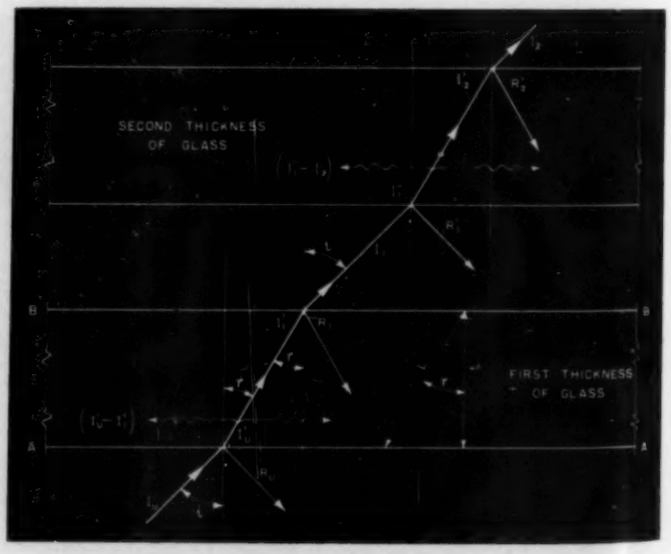


Fig. 8—Path of ray of sunlight through single and double glass, showing energy transmitted, reflected, and absorbed

The problem here is to find the radiant energy, I_1, of the ray which finally emerges from the first thickness of glass. Inasmuch as only a single thickness of glass is being discussed here, the second thickness illustrated in Fig. 8 is to be ignored for the present. The energy reflected and transmitted at the boundary surface *A-A* is related to the incident energy by

$$I'_a = I_a - R_a \quad \text{.....[a]}$$

$$I'_a = I_a \left(1 - \frac{R_a}{I_a}\right)$$

Let $$P_F = \frac{R_a}{I_a}$$

$$I'_a = I_a\ (1 - P_F) \quad \text{.....[b]}$$

P_F = per cent of incident light reflected from a single boundary surface.

The value of P_F is computed by means of Fresnel's formula which is given in Appendix 7.

The value of P_F above is not the total light reflected, as further reflection takes place at the inner surface of the glass when the ray passes the surface *B-B* from the glass into the air. This is also illustrated in Fig. 8.

At the inner boundary surface *B-B*,

$$I_1 = I'_1 - R_1 \quad \text{.....[c]}$$

$$I_1 = I'_1 \left(1 - \frac{R_1}{I'_1}\right) \quad \text{.....[d]}$$

Now for the same angles of incidence and refraction, it is evident from Fresnel's formula (see Appendix 7) that

$$P_F = \frac{R_a}{I_a} = \frac{R_1}{I'_1} \quad \text{.....[e]}$$

Therefore,

$$I_1 = I'_1\ (1 - P_F) \quad \text{.....[f]}$$

$$I'_1 = \frac{I_1}{1 - P_F} \quad \text{.....[g]}$$

Now a third formula is needed to relate I'_1 and I'_a. Of the incident energy I_a, the energy I'_a (see Fig. 8) passes the boundary. If there were no absorption of energy by the glass itself, the energy I'_a would impinge on the inner boundary *B-B*. Because of the absorption of energy by the glass itself, the energy incident at the inner boundary *B-B* is reduced to I'_1, where[a],

$$\frac{I'_1}{I'_a} = e^{-kL} \quad \text{.....[h]}$$

In this formula k is an experimentally determined constant whose value depends upon the characteristics of the substance through which the ray is passing. The value of L, the length of the path that the ray follows through the glass, can be computed by means of the following formula, whose derivation is obvious from the geometry of Fig. 8,

$$L = \frac{t}{\cos r} \quad \text{.....[j]}$$

t = thickness of glass;
r = angle of refraction.

Substitute [b] and [g] into [h], yielding

$$\frac{I_1}{I_a} = (1 - P_F)^2\ e^{-kL} \quad \text{.....[k]}$$

Now I_1/I_a is the fraction of the incident energy that is transmitted through the glass as radiant energy. Let

$$P_{T1} = \frac{I_1}{I_a} \quad \text{.....[l]}$$

Therefore,

$$P_{T1} = (1 - P_F)^2\ e^{-kL}$$

[a] Industrial Heat Transfer, Schack, Goldschmidt, and Partridge, p. 183.

Appendix 4—Derivation of Equation 8

$$P_{A1} = (1 - P_F)\ (1 - e^{-kL})$$

Referring to Fig. 8, the radiant energy absorbed by the single glass is $(I'_a - I'_1)$.

From [b] of Appendix 3,

$$I'_a = I_a\ (1 - P_F) \quad \text{.....[a]}$$

From [g] of Appendix 3,

$$I'_1 = \frac{I_1}{1 - P_F} \quad \text{.....[b]}$$

Subtract [b] from [a],

$$I'_a - I'_1 = I_a\ (1 - P_F) - \frac{I_1}{1 - P_F}$$

$$\frac{I'_a - I'_1}{I_a} = (1 - P_F) - \frac{I_1}{I_a}\,\frac{1}{1 - P_F} \quad \text{.....[c]}$$

Eliminate I_1/I_a from [c] by means of [k] of Appendix 3, resulting in

$$\frac{I'_a - I'_1}{I_a} = (1 - P_F) - (1 - P_F)e^{-kL}$$

$$= (1 - P_F)\ (1 - e^{-kL}) \quad \text{.....[d]}$$

Now $\dfrac{I'_a - I'_1}{I_a}$ is the fraction of the incident energy that is absorbed by the glass and converted into heat, or

$$P_{A1} = \frac{I'_a - I'_1}{I_a} \quad \text{.....[e]}$$

Therefore,

$$P_{A1} = (1 - P_F)\ (1 - e^{-kL})$$

Appendix 5—Derivation of Equation 9

$$P_{T2} = P^2_{T1}$$

The ray of light travels through parallel plates of glass of equal thickness. As before, I_a is the energy incident to the first plate of glass, and I_1 is the energy leaving the first glass and entering the second glass. I_2 is the energy of the ray of light leaving the second thickness of glass. The problem here is to find the radiant energy I_2 of the ray that finally emerges from the second glass.

Evidently an equation analogous to [k] of Appendix 3 is also applicable to the second plate of glass alone, or

$$\frac{I_2}{I_1} = (1 - P_F)^2 e^{-kL} \quad \text{.....[a]}$$

Also from [k] of Appendix 3,

$$\frac{I_1}{I_a} = (1 - P_F)^2 e^{-kL} \quad \text{.....[b]}$$

Multiplying [a] and [b],

$$\frac{I_2}{I_a} = (1 - P_F)^4 e^{-2kL} \quad \text{.....[c]}$$

But, squaring Equation 7 of the text,

$$P^2_{T1} = (1 - P_F)^4 e^{-2kL}$$

Therefore, $$\frac{I_2}{I_a} = P^2_{T1} \quad \text{.....[d]}$$

Now (I_2/I_a) is the fraction of the incident energy that is transmitted through the two plates of glass. Let

$$P_{T2} = \frac{I_2}{I_a}$$

Therefore, $P_{T2} = P^2_{T1}$

[To be continued]

Brass Piping Protects Product at Vermont Maple Sugar Plant

By **Augustus Conlin* (with Francis A. Westbrook)**

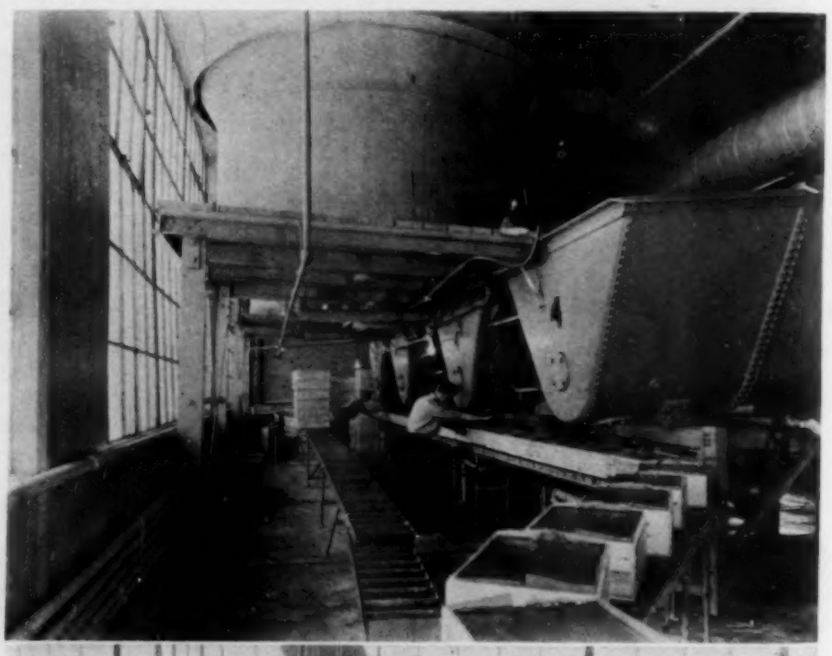
BRASS piping for the handling of hot maple syrup is an important item of equipment at the plant of the Cary Maple Sugar Co., St. Johnsbury, Vermont. Brass is used both from the standpoint of maintenance and for protecting the quality of the product.

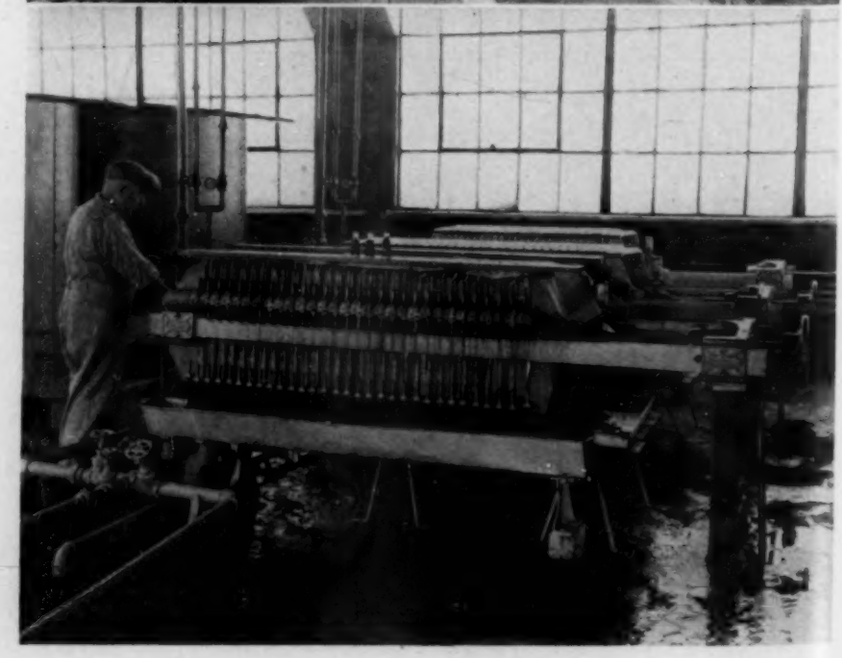
The extent to which syrup is handled in piping will be appreciated from the following brief description. The material is received in large drums from producers in about 10 different states. It is dumped into tanks on the first floor and pumped to large storage tanks on the fourth floor. From these the syrup may descend by gravity to the cookers on the second floor where it is boiled down for the making of sugar, or it may go down to the third floor to a vacuum tank where it is standardized as to density. It is in connection with the latter that the most interesting use of brass piping is to be found.

Adjacent to the 500 gal standardization tank there are three 2000 gal storage tanks and one of 1000 gal. One of the 2000 gal tanks is next to the 500 gal tank and is used for the temporary storage of unfiltered syrup; that is, after the syrup has been standardized it must be filtered before bottling. It is sometimes pumped direct to the plate and frame filters on the floor above, or if this is not convenient it can be pumped into the storage tank first and then to the filter later on. After filtering it passes by gravity to one of three storage tanks for filtered syrup. When it is to be used for bottling it is forced by air pressure to a small tank near the ceiling and close to the filling machine, to which it descends by gravity. In order to be free flowing and to insure sterilization its temperature must be around 175 F.

Obviously, considerable piping is called for and several pumps have been installed under the tanks; these are of the single cylinder steam driven type. They are arranged for pumping from the standardization tank to the filter or to the unfiltered storage tank, or from the latter to the filter. The pumps have brass linings. Pipes and pumps have also been installed to provide for flushing the tanks and syrup pipes with hot water. Such cleaning is essential

[Concluded on p. 579]





Above—Cooker for boiling down syrup for the production of maple sugar. Copper ducts are used for carrying off the vapor. . . . Below—A view of the filters, showing some of the brass piping

*Chemist, Cary Maple Sugar Co., Inc.

Motors in the Heating Plant

By Kalman Steiner* and Fred Ravnsbeck†

Not a technical treatise on electric motors, but a simplified description of the types ordinarily used with plant and building heating systems is given here. . . . The authors are primarily heating men, write from the viewpoint of the operating engineer concerned with operation and maintenance of the motors which are a part of the heating system

THE review given in July[1] of d-c and polyphase a-c motors covered the great majority of boiler room motors. There remains to be discussed one group of a-c motors, known as single phase motors. A brief consideration of them is justified because there are heating plants in commercial buildings, apartments, small hotels and public buildings where only single phase current is available. In addition, many single phase motors are used to drive electric fans of the common propeller blade type.

Single Phase Motors

The earliest form of single phase was probably the **split phase** motor, and it is still frequently used. Its rotor was described with the squirrel cage and capacitor rotors, as all are similar. Many manufacturers who offer squirrel cage and split phase motors in the same sizes and capacities make the rotors of the two types interchangeable. From the viewpoint of maintenance service, this is of some advantage, as not infrequently a temporary split phase stator is slipped over a squirrel cage rotor to keep a machine running while the regular stator is being rewound. Or again, during construction of a new building, the stoker or oil burner is required to deliver temporary heat for permitting construction operations during cold weather before the building is completed and before the public service company has provided three phase current. In such cases, the interchangeability of the two kinds of stators makes it possible to operate the boiler under mechanical firing with but a very simple change in the motor.

The stator of the split phase consists of two windings, starting and running. The starting winding is very light and is able to carry current for a few seconds before it overheats, and in overheating it will damage both itself and the running winding. The starting winding circuit must be opened by a centrifugal device the instant proper speed is attained. Various types of centrifugal devices are employed, most of which are entirely satisfactory at speeds of 1800 or less but may not be at 3600 rpm. Some of them are delicate and offer operating difficulties. Failure of the centrifugal switch to open usually results in both windings burning out, or at least the starting winding is damaged. Perhaps the inherent fault of the centrifugal switch is that fairly expert workmanship is required in removing and replacing the motor or its end bells, and in the routine servicing of small motors or the appliances which they drive that expertness is sometimes lacking.

Another objection to the centrifugal switch is that it sometimes interferes with the logical arrangement of the motor with respect to the rest of the mechanism, since accessibility must be permitted to the end bell and the centrifugal switch for servicing, and in the case of horizontal rotary oil burners the switch interferes with the passage of an oil feeding arrangement through the motor shaft, or occupies the space around the shaft which should be reserved for oil seals and other parts. But the final and strongest objection to split phase motors is the one raised by power companies themselves; the starting current is very heavy. In certain localities (Chicago, for instance) the public service companies have rulings against the use of split phase motors except in very small sizes, less than 1/6 hp. That is, certain maximum allowable starting currents are established for each rating, and split phase motors usually fail to comply with these.

The second type of single phase motor, the **repulsion induction,** starts by electrical repulsion and then runs by induction, just as do all other types of a-c motors except the synchronous. Starting is accomplished by placing a winding on the rotor, in which a current is induced from the current applied to the stator. The electrical nature of the motor requires that after it reaches speed, a short circuiting device must be brought into contact with all segments of the commutator. The latter, with a set of brushes, is required to permit the motor to start. Generally, too, the brushes are lifted away from the commutator when not needed. All this means a mechanical system of some complexity, usually operated by a centrifugal governor. Again it can be said that here is a motor which gives a high degree of satisfactory performance at low and moderate speeds, but which exhibits objectionable behavior at 3600 rpm. Particularly objectionable from the standpoint of horizontal rotary oil burner application is the need for a rotor winding, for this necessitates a two piece screwed shaft, instead of the single piece shaft which is possible with other types of a-c motors. In the cases of both oil burners and stokers, grime and dust soon coat the commutator and necessitate periodic cleaning.

Last in the single phase motor group is the **capacitor**

*Mid-West Heat Service Co.
†Ace Engineering Co.
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[1] Heating, Piping and Air Conditioning, July, 1938, pp. 441-444.

motor. Most recent developments in the single phase field have been with this motor, although its theory is not new.

This motor overcomes many of the objectionable features inherent in the split phase and repulsion induction; the high starting current of the former and the wound rotor and brushes for the latter are not found in the capacitor. In addition, the rotor (as with the split phase) can be made identical to and interchangeable with the polyphase rotor. The capacitor motor has a starting winding, but has in addition a capacitor which is not integral with either the stator or rotor. It may be housed in the stator casing or upon it, or may be contained within a separate box that need not be physically attached to the motor at all, but may be hung or supported at any convenient point, such as the panel board which carries the motor starting relays, controls, etc. The capacitor housing will also contain the magnetic relay which cuts out the starting winding when the motor is up to speed, if that style of cutout is used. The current that energizes the magnetic relay to accomplish this is induced by the rotating motor, and no centrifugal or other mechanism is required on the motor itself. But the general practice is to use a centrifugal switch up to ½ hp.

As regularly made, the capacitor motor is a constant speed machine, like other standard a-c types. Many of its common applications, however, can use speed control to excellent advantage. Typical of such are electric fans. For this purpose a rather unusual method is resorted to. Ordinarily, the speed of an induction motor is independent of the voltage. But with the capacitor, it is possible to secure from a transformer a voltage that will cause an excessive amount of *slip*, which in any induction motor is the ratio or percentage expressing the relationship between actual motor speed and the synchronous speed. In such motors the voltage is increased for starting and then reduced by having a centrifugal switch change the connection after the motor is up to speed.

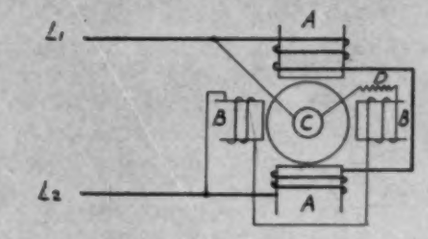

Fig. 1

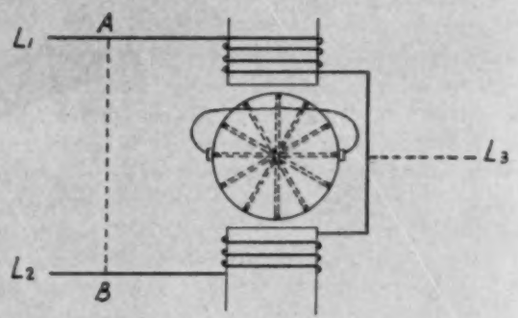

Fig. 2

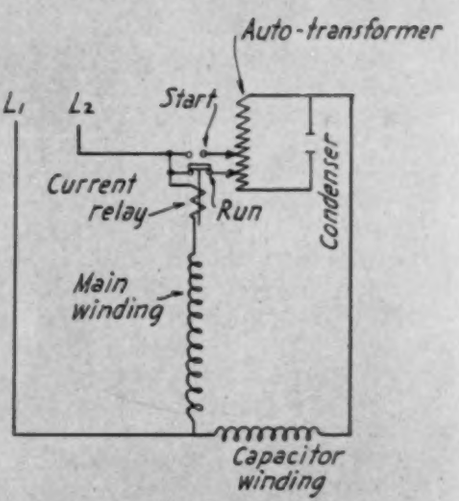

Fig. 3

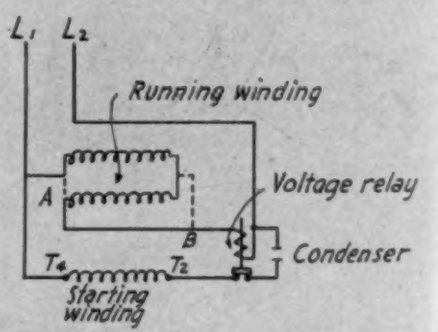

Fig. 4

Fig. 1—Diagram of split phase motor. *A*, *A* are the main poles; *B*, *B* are the starting poles; *C* is the centrifugal switch which opens the starting winding as the motor comes to speed; *D* is a resistance in the starting winding

Fig. 2—Diagram of repulsion-induction motor. As motor starts, the brushes successively make a circuit around one of the rotor conductors. When the rotor turns fast enough to operate a centrifugal device, a short-circuiting mechanism makes a single contact between all the rotor conductors. The line voltage across L_1 and L_2 is 220; by connecting *A* to *B*, and changing L_1 to become L_3, the motor runs on 110 volts

Fig. 3—Diagram of capacitor start, capacitor run motor. When the motor is at rest, L_2 connects to "run" position in current relay. At instant of starting, heavy inrush current activates the current relay, connecting L_2 to "start" position. When motor reaches ¾ speed, diminishing current drops out relay, reconnecting L_2 to "run" position. The auto-transformer, besides serving during the starting period, may also be equipped with a manual setting device to induce different voltages, which in turn will result in different motor speeds

Fig. 4—Diagram of capacitor start, induction run motor. When the relay pulls in, because of the voltage changes caused by the motor approaching speed, the starting winding is opened. As shown, the connections are for a line voltage of 220 volts. By merely making the connections indicated by the broken lines, and removing the connection between *A* and *B*, the motor will run on line voltage of 110 volts

The transformer will be built in with the starting condenser. Usually, the rotor of an adjustable speed motor is of slightly different construction from standard constant speed forms.

Propeller type fans—such as are often used to exhaust air from a large public room or kitchen—do not need a high starting torque, since the load builds up with the speed. Here, for constant speed work, split phase motors will do. But when speed control is desired, a variable speed type of capacitor must be used. Pumps, oil burners, and stokers, which do not require motor speeds above 1750 rpm, can use repulsion induction motors, as such applications are almost universally constant speed. But at higher speeds, even in constant speed applications, the capacitor motor can be more profitably applied. One of the largest electrical manufacturers in the country has discontinued the manufacture of split phase motors, preferring to cover the entire range of single phase motor work with the repulsion induction and capacitor types.

Starting Induction Motors

Induction motors up to 10 hp can generally be started across the line; that is, the main line contacts to the motor can be closed and full line voltage applied to the motor while it is at rest. Such starting is most often done by having a solenoid coil pull in an iron bar which will in turn carry with it the motor starting contacts. The typical across-the-line starter will have a pushbutton station, if the motor is to be started manually, with one button for starting and another for stopping, the contacts being in the coil circuit. For automatic starting of pumps, burners, or stokers, the coil circuit is closed through some automatic control device.

The comparatively high inrush current when starting induction motors above 10 hp prevents throwing them across the line. There are two methods for starting large squirrel cage motors, both serving to reduce the starting voltage. The first consists merely in placing sufficient resistance in the line to bring the voltage down to a value which will not cause a heavy current flow. The second is to use auto-transformers. By this last method the reduction in voltage does not entail a loss in current through the resistance, as does the first. The common compensator type of starter, as the auto-transformer starter is called, has a manually operated two position switch. The switch is first thrown into the position which will enable the auto-transformer to start the motor at reduced voltage; the switch is then thrown to the running position and the motor is across the line. The running position throws into the circuit also overload protective devices and low voltage release.

Motor Frames

Motor frames are available in a variety of styles. The nature of the frame is indicated by the name applied to it. Customary styles of frames are open, semi-enclosed, totally enclosed, enclosed ventilated, enclosed fan cooled, splash proof, explosion proof, and so on. The open frame is generally employed unless the conditions of operation or installation require one of the other special types.

Slip

In closing, the authors take a few lines to amplify a subject which was given slight attention in a previous paragraph—the term *slip*. When an alternating current supply is connected to the stator of an induction motor, a rotating field is set up. As this field revolves around the rotor, it cuts the conductors of the rotor, and thereby induces in the rotor winding another electric current. The effect of the two currents is to cause the rotor to turn. There are, therefore, actually two rotations to consider—that of the rotating field set up in the stator, and the rotation of the rotor. The former is called the *synchronous speed*, and it is this speed which is referred to in the formula given in the July article.[1] The latter is the actual motor rpm.

It is apparent that should the actual speed equal the synchronous speed, there would no longer be an inducing force, for it is the cutting of the rotor conductors by the rotating field that induces the rotor current. When the two are not in relative motion to each other, induction ceases. The difference between synchronous speed and rpm is called the *slip*. The greater the load, the greater the slip. That is, as load causes the motor to slow down, the slip is of course greater. But as has been brought out, induction motor speed regulation is held within 5 per cent, which means that the slip does not increase more than that percentage between no load and full load.

Maple Sugar Plant—

[Concluded from p. 576]

as stale syrup ferments and will contaminate the packaged product.

The 1 in. hot water piping in the bottling department is also of brass, and brass is being used for all replacements throughout the factory for the hot water lines. The brass piping for the hot syrup is 1½ and 2 in. in size.

Copper Ducts Remove Vapor

Another interesting feature of the plant is the system of copper ducts for carrying off the vapors rising from the kettles in which syrup is cooked in the process of making maple sugar, which is used in the tobacco industry. There are two cookers on each side of the room and each has a separate line, the two joining near the exhaust fan. The first section is 12 in. in diameter and from the second cooker to the blower 16 in. in diameter. The outlet from each cooker is 10 in. and when the two 16 in. sections join the diameter is 24 in. which is the size of the inlet to the blower. This arrangement prevents steam and the accompanying fumes from the cooking process from entering the room and keeps the atmosphere clear at all times. This is particularly important during the extremely cold days which occur off and on in winter, for the condensation of moisture on the outside walls would be a serious matter at such times if an excessive amount of steam were present in the room.

Eliminating Fire Hazards in Air Conditioning Work

By H. C. Murphy*

ELIMINATION of possible fire hazards in ventilating systems has been studied by various groups for a number of years. As early as 1899, the National Fire Protection Association discussed methods of securing maximum safety in the installation and operation of blower and exhaust systems.

In 1900, the N.F.P.A. committee on blower systems issued its first recommendation on this subject. These regulations have from time to time been revised to meet new developments and conditions. The complete regulations are contained in a booklet which may be obtained from the National Fire Protection Association or the National Board of Fire Underwriters by applying for their Pamphlet No. 90.

At the annual convention of the N.F.P.A. recently held in Atlantic City, the following amendments to the regulations as contained in Pamphlet No. 90 were adopted. These revisions are the result of careful study and investigation of present conditions by the committee, and compliance with these rules will be of undoubted benefit to all concerned.

PART I, AIR CONDITIONING AND VENTILATING SYSTEMS IN OTHER THAN RESIDENCES

Add new paragraph 114A, as follows:

114A. Combustible coverings or combustible exterior insulations to ducts are not recommended and should not be used generally, particularly in concealed locations of non-fire-resistive construction, unless covered with a cement or gypsum plaster coating of ¼ in. thickness. On ducts with entering air temperatures exceeding 175 F combustible coverings shall not be used.

Repair work involving the use of torches should not be undertaken on ducts covered with combustible material until such material has been removed from that portion of the duct.

Add new paragraph 117, as follows:

117. Return ducts, other than vertical, shall be so constructed that the interior is accessible to facilitate the cleaning of possible accumulations of dust and combustible material in them. Cleanout openings at approximately 20 ft intervals shall be provided where the ducts are of such size that they may not readily be entered to accomplish the cleaning.

Supply ducts, other than vertical, shall conform to the above regulations for return ducts, unless all of the supply air passes through either water spray or filters.

Revise paragraph 135 to read:

135. Where ducts pass through floors, approved fire dampers shall be provided at each outlet, inlet, and branch opening in such main vertical duct. Where such ducts serve only one floor, only one fire damper shall be required in the main supply duct from the unit and one in the return.

Revise paragraph 142 to read:

142. Air inlet and outlet openings shall be located at least 3 in. above the floor, except that protected floor inlets may be permitted, under seats, in theaters. When located less than 7 ft above the floor, inlet and outlet openings shall be protected by a substantial grille or screen, through the openings in which a half-inch sphere will not pass.

Revise paragraph 151 to read:

151. Combustible air filters should not be used unless they are in themselves, or by treatment, sufficiently fire-resistive so that fire spreading over the filter when loaded with dust and under operating conditions will not be materially fed by burning of the filter itself.

The installation of approved automatic extinguishing equipment employing water, gas or other suitable means is recommended in the enclosure of the air conditioning system to protect against combustion of material that may accumulate, except that protection is not required in the unit or cabinet type of air conditioning equipment with blower capacity not exceeding 20,000 cfm and where the unit supplies only one floor area or a portion thereof. Where sprinklers are installed, suitable provision should be made for drainage. In buildings not equipped with automatic sprinklers, the water supply may be taken from the house piping, if the supply is adequate for the purpose.

Add new sentence to paragraph 181, making it read:

181. Electric wiring and equipment shall be installed in accordance with the National Electrical Code. Lamps within the enclosure of the conditioning system shall be enclosed in fixtures of the marine (vapor tight) type.

Note—It is recognized that in certain occupancies there is a potential panic and smoke damage hazard. For such conditions it is recommended that consideration be given to the development and application of suitable smoke detectors to automatically shut down the system and cause an alarm signal.

*Vice-president, American Air Filter Co. Member of Committee on Blower Systems, National Fire Protection Association. Member of Board of Consulting and Contributing Editors.

Fuel Consumption and Color of Radiator

An impression has been circulated that painting radiators different colors affects building fuel consumption. The basis is the finding that a black or dark radiator emits more heat per square foot of surface than a radiator painted a lighter color.

"This finding is perfectly logical and true," says Allen J. Johnson, director of the Anthracite Industries Laboratory, "but the fallacy of attempting to apply this to an interpretation of coal consumption is that if a room is adequately heated, the amount of heat it consumes is independent of the size or efficiency of the radiators. The heat is used not for heating the radiator but rather the radiator acts as a medium for holding the heat until it can be discharged into the room. Therefore, the question of fuel consumption becomes not one of the radiator's efficiency but rather one of outdoor weather, building construction, and the extent to which the building is heated."

Welding High Pressure, High Temperature Piping

By W. J. Angus* and C. A. Kelting†

The story of the design and installation of the high pressure, high temperature piping for Consolidated Edison's Waterside Station is continued with this description of the details of the welding, stress relieving, support of the piping, and weld examination

GENERAL design of the piping involved in the installation of the 1200-1400 lb, 900 F topping turbines and their associated boilers at the Waterside Station of the Consolidated Edison Co. of New York Inc. was discussed last month.[1] Details of the welding procedure, stress relieving, examination of the welds, and the support of the piping are considered this month.

The welding of piping joints, both in the shop and the field, for the two units was performed by welding operators qualified under the rules of the Travelers Indemnity Co., which follow those of the Code for Pressure Piping with some extra requirements, one of which included the making of a satisfactory branch connection.

Weld Details and Backing Rings

Welding was performed by the direct current electric arc method using covered electrodes. These electrodes or welding rods were of a steel having approximately the same composition as the pipe, valve, or fitting being welded and in the case of welding any component part made of carbon molybdenum alloy steel, the molybdenum was contained in the proper amount in the coating or covering of the suitable carbon steel welding rods.

All welded joints in piping of nominal sizes 2 in. and larger are of the butt welded design, using a backing ring to prevent the formation of icicles or weld splatter inside the pipe. These backing rings are ⅛ or $\frac{3}{16}$ in. thick in some cases for larger pipe sizes, and are plain, flat split rings 1 in. wide without spacing ribs or grooves. They are of steel composition, approximately the same as that part of the piping with which they are used. The design of these joints is shown in Figs. 1 and 2.

A. S. A. sectional committee B16 is formulating a dimensional standard for welding fittings and after some controversy has proposed that the welding bevel recommended for standardization be as will appear in the forthcoming revision to A. S. A. Standard B16e for steel flanges and steel flanged fittings. This in brief calls for a 37½ deg straight bevel, with a plus or minus tolerance of 2½ deg and a $\frac{1}{16}$ in. land with a plus or minus tolerance of $\frac{1}{32}$ in. for wall thicknesses of welding ends $\frac{3}{16}$ to ¾ in. inclusive. That recommended for wall thicknesses greater than ¾ in. calls for a 20 deg (± 2½ deg) U bevel with a $\frac{3}{16}$ in. radius at the bottom and a $\frac{1}{16}$ in. (± $\frac{1}{32}$ in.) land.

Joints on piping smaller than 2 in., where welded, were of the fillet strength welded type using sleeves or plain couplings shown by Fig. 3 for pipe-to-pipe joints and socket welding end fittings and valves for pipe-to-fitting or valve joints as shown by Fig. 4. This form of joint was found to be serviceable for such sizes and eliminates the need of a backing ring, which is objectionable in small size lines because of the obstruction it causes.

The dimension for bore of the sockets for Fig. 4 is the same as *D* of Fig. 3.

Welding ends of all pipe, fittings, and valves were machined, and for the butt welded joints suitably bevelled, dependent upon pipe wall thickness as mentioned. It was found that in order to facilitate the depositing of weld metal in the heavy wall joints to obtain thorough fusion at the bottom of the joint the 26 deg U form of bevel design shown by Fig. 2 proved quite satisfactory.

Tests indicated it advisable, in order to facilitate welding and produce a more ductile weld, that the parts to be joined when welding carbon molybdenum alloy steel be preheated. Therefore, all joints on the main steam piping and the carbon molybdenum valve joints on the boiler-feed and other lines were preheated to about 400 to 600 F before welding. Such preheating on the first unit was accomplished by torches, although as shown by Fig. 5 the joint at the turbine throttle made by the turbine manufacturer was preheated by means of a special electric resistance wire heating element wrapped around the joining parts and covered by insulation to retain the heat and protect the welding operator.

Experience gained during the welding of carbon molybdenum alloy steel on the first unit revealed a further benefit to be gained by preheating and the maintaining of such heat until welding had been completed. Although carbon molybdenum steel is considered not to have appreciable air hardening tendencies, evidences of some were produced and cracks developed in the weld when the partially completed or finished joint was allowed to cool down before being stress relieved. To prevent this at first, the preheat temperature was maintained during welding by the heat of welding, employing two operators on the 12 in. joint at the same time; aside from the cleaning of beads, the welding was con-

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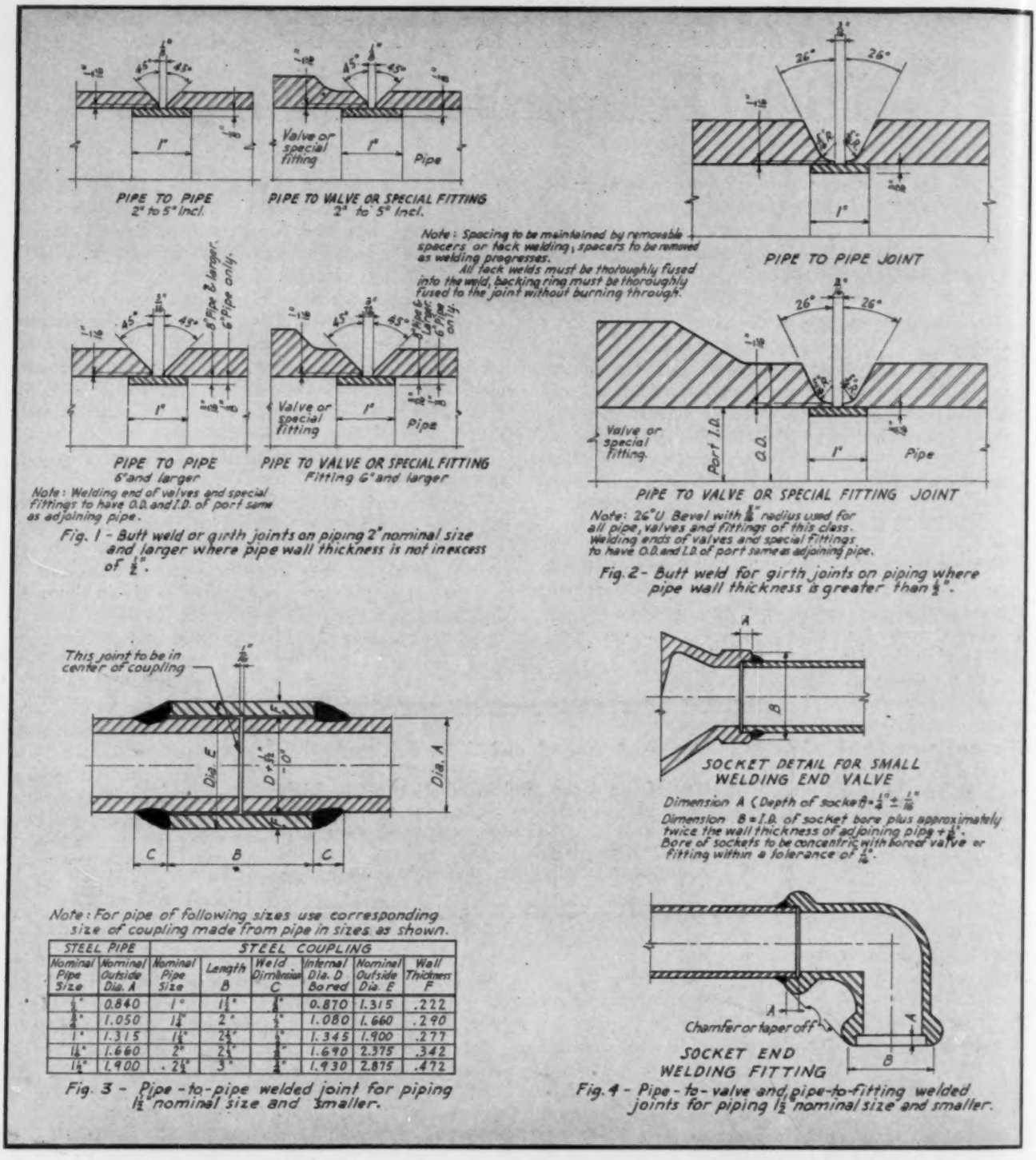
†Assistant Division Engineer, Consolidated Edison Co. of New York Inc.

[1] Heating, Piping and Air Conditioning, August, 1938, pp. 512-514.


PIPE TO PIPE
2" to 5" Incl.
PIPE TO VALVE OR SPECIAL FITTING
2" to 5" Incl.
Valve or special fitting
Pipe
PIPE TO PIPE
6" and larger
PIPE TO VALVE OR SPECIAL FITTING
Fitting 6" and larger
Note: Welding end of valves and special fittings to have O.D. and I.D. of port same as adjoining pipe.
Fig. 1 - Butt weld or girth joints on piping 2" nominal size and larger where pipe wall thickness is not in excess of 1/2".
Note: Spacing to be maintained by removable spacers or tack welding, spacers to be removed as welding progresses.
All tack welds must be thoroughly fused into the weld, backing ring must be thoroughly fused to the joint without burning through.
PIPE TO PIPE JOINT
PIPE TO VALVE OR SPECIAL FITTING JOINT
Port I.D.
O.D.
Note: 26° U Bevel with 3/16" radius used for all pipe, valves and fittings of this class. Welding ends of valves and special fittings to have O.D. and I.D. of port same as adjoining pipe.
Fig. 2 - Butt weld for girth joints on piping where pipe wall thickness is greater than 1/2".
This joint to be in center of coupling
Dia. E
Dia. A
C
B
C
Note: For pipe of following sizes use corresponding size of coupling made from pipe in sizes as shown.
SOCKET DETAIL FOR SMALL WELDING END VALVE
Dimension A (Depth of socket) = 1/4" ± 1/16"
Dimension B = I.D. of socket bore plus approximately twice the wall thickness of adjoining pipe + 1/16". Bore of sockets to be concentric with bored valve or fitting within a tolerance of 1/16".
Chamfer or taper off
SOCKET END WELDING FITTING
Fig. 3 - Pipe-to-pipe welded joint for piping 1 1/2" nominal size and smaller.
Fig. 4 - Pipe-to-valve and pipe-to-fitting welded joints for piping 1 1/2" nominal size and smaller.

STEEL PIPE		STEEL COUPLING					
Nominal Pipe Size	Nominal Outside Dia. A	Nominal Pipe Size	Length B	Weld Dimension C	Internal Dia. D Bored	Nominal Outside Dia. E	Wall Thickness F
1/8"	0.840	1"	1 1/2"	1/4"	0.870	1.315	.222
1/4"	1.050	1 1/4"	2"	1/4"	1.080	1.660	.290
1"	1.315	1 1/2"	2 1/2"	1/4"	1.345	1.900	.277
1 1/4"	1.660	2"	2 1/2"	1/4"	1.690	2.375	.342
1 1/2"	1.900	2 1/2"	3"	1/4"	1.930	2.875	.472

tinuous until the entire joint had been completed. Immediately following this, the joint was stress relieved.

On the second unit, advantage was taken of the favorable characteristics offered by wrapping a heavy stranded copper wire cable, with asbestos insulation between the turns, around the parts joined, as shown by Fig. 6. This winding, supplied with alternating current, acts as the primary of a transformer with the pipe as its short circuited secondary and heat is generated within the piping parts thus enclosed through induction. Preheating was established by this means but due to the effects of the alternating current magnetic field upon the welding arc, the current on the winding had to be cut off at such times when welding was in progress.

Two operators welded on the joint at the same time at opposite sides to maintain the heat but when a bead was completed, the current on the winding was put on again to maintain the joint at temperature during the cleaning and chipping of the beads. In this way, where a joint could not be completed at the time, the parts were maintained at temperature overnight or over a week end continually until such time when welding was resumed. When the weld was finished the temperature was raised by the same equipment for the stress reliev-

ing of the joint. Following this procedure enabled us to eliminate the formation of cracks in the weld and produced very satisfactory results.

All welding was done by careful application of the first bead or layer, using a smaller diameter welding rod at first with precaution taken not to burn through the backing ring but at the same time to produce thorough penetration and fusion at the bottom of the joint. To obtain a satisfactory sound weld, a greater number of thin beads or layers were laid in rather than to deposit a fewer number of heavier ones. Company specifications called for not less than one bead per ⅛ in. of pipe wall thickness and prohibited the use of electrodes larger than $\frac{3}{16}$ in. diameter.

Stress Relieving

In the course of welding, stresses are set up in the weld and for a distance back in the parent metal. These internal locked up stresses are the cause of welding cracks which develop unless such stresses are relieved. It is especially important to relieve these stresses in steels of the alloys which have any tendency to air-harden and produce brittleness.

Heat treatment causing refinement of the grain after welding will insure the most satisfactory results in obtaining high strength with great ductility, resiliency, and resistance to shock. Such, therefore, was the treatment given in the stress relieving of welds by allowing the welds to partially cool to a black heat and then reheating through the critical range, with care taken not to heat above the upper limit in order to prevent grain growth.

While of special importance with alloy steels, it is also felt to be valuable to stress relieve carbon steel welded joints in lines under severe or important service. Consequently, all welded joints of 2½ in. nominal size and larger on the main steam, the boilerfeed discharge, and the 200 lb exhaust and station steam tie lines were stress relieved. This was accomplished on field welded joints by means of an electric inductance type of stress relieving furnace as shown by Fig. 7, as well as by the equipment used for carbon molybdenum welding previously mentioned.

The pipe was heated uniformly until a band on the pipe circumference of a width containing the weld and 1 in. of parent metal on each side of the weld was brought up slowly to a temperature between 1100 to 1200 F. This heat was held for a period of one hour per inch of pipe wall thickness in the case of carbon steel and two hours per inch of thickness for carbon molybdenum steel. Then the joint was allowed to cool slowly by gradually tapering off the current, followed by cooling in the atmosphere.

Welds made in the piping fabricator's shop were stress relieved in this fashion also, and in some cases where large fuel fired furnaces were available the entire piece was stress relieved in the furnace.

The load on the inductive stress relieving coils was controlled by a portable panel and transformer. A continuous temperature recorder on this panel made a record of the temperatures transmitted from an electric pyrometer located at the weld in each case. With this

Fig. 5 (top)—Preheating and annealing arrangement for joint at turbine throttle, first unit. . . Fig. 6 (center)—Induction method of heating and annealing used on second unit. . . Fig. 7 (bottom)—Stress reliever on welded heater connection

equipment and these methods available the work progressed quite rapidly throughout the installation.

Valve Design and Installation

The statement made last month to the effect that carbon molybdenum alloy steel is as good at 900 F as carbon steel is at 750 F was found to be the basis of design on the part of valve manufacturers for valves

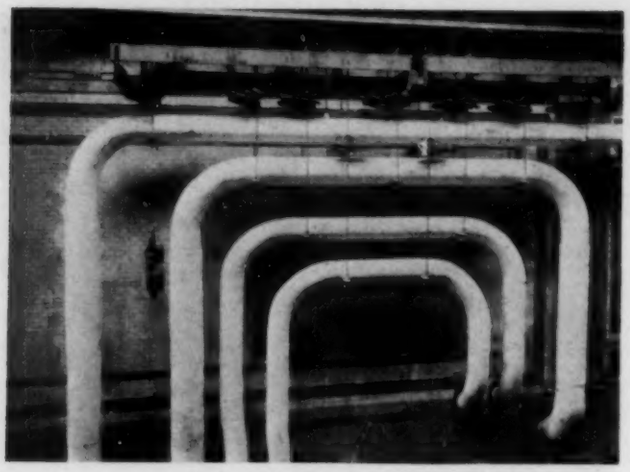


Fig. 8—Main steam and 200 lb bleed steam piping, showing spring supports

suitable for this service. It enabled them to recommend and offer steel valves, rated for the various pressures at a maximum temperature of 750 F for carbon steel, as being suitable for such respective pressures at 900 F if made of carbon molybdenum steel. After careful investigation of the details involved, this recommendation was accepted and consequently for our 1200-1400 lb per sq in., 900 F main steam service, valves of the 1500 lb standard series were used with bodies and bonnets of carbon molybdenum steel.

In general all steel valves were welded directly in the line in the course of erection, the purpose being to eliminate line joint trouble.

The main steam line valves for the first unit have nitrided seating, while the steel valves in the boilerfeed and the other lower temperature steam services have one or the other of the stainless steel combinations. Some miscellaneous high pressure, high temperature valves scattered throughout have stellite seating by virtue of special recommendation of some of the equipment manufacturers. However, in addition to those several miscellaneous valves, for the second unit installation the main steam line valves are also with stellite seating as there has been a growing tendency to favor that material for high temperature service.

This material is a non-ferrous alloy of cobalt, chromium, and tungsten with exceptional resistance to abrasion. Combined with its inherent property of uniform hardness is its so-called "red hardness," which is the measure of hardness at red heat. This is a highly favorable quality which accounts for its superiority in resistance to wear and to seizing and cutting of the valve seats under severe service, even at high temperatures.

Of the three grades available, the No. 6 was used on our valves so trimmed. It is not quite as hard as the No. 1 or No. 12 grades but produces a Brinnell of about 400, which is quite satisfactory for the service. Being much stronger and tougher than the other two grades, it is not expected to check or chip as easily. Its coefficient of expansion closely approximates the carbon molybdenum alloy steel which may be used for the wedges or discs and the body seat rings, and therefore can be applied quite successfully to these for surfacing—even to large areas.

Support of the Piping

The high rate of pipe expansion combined with space limitations, high material and installation costs presented the difficult problem of designing an economical pipe run which would be capable of absorbing the expansion forces at unit stresses suitable to the severe service conditions of the line. The three dimensional expansions between superheater outlet and turbine inlet varied from as little as 2 in. in a vertical direction to as much as 13 in. in the direction of the longitudinal axis of the pipe. The solution finally adopted resulted in a combined longitudinal stress due to bending and internal pressure of 9500 lb per sq in., including the effect of pipe flattening in the curved portions of the pipe. The lines were erected from both ends toward the middle, allowing for the insertion of field pieces which were shortened and offset to allow for cold springing in three directions to the extent of 50 per cent of the corresponding expansions. Conforming to the requirements of the A.S.A. Code for Pressure Piping no reduction in pipe stress was allowed due to the beneficial effect of cold springing.

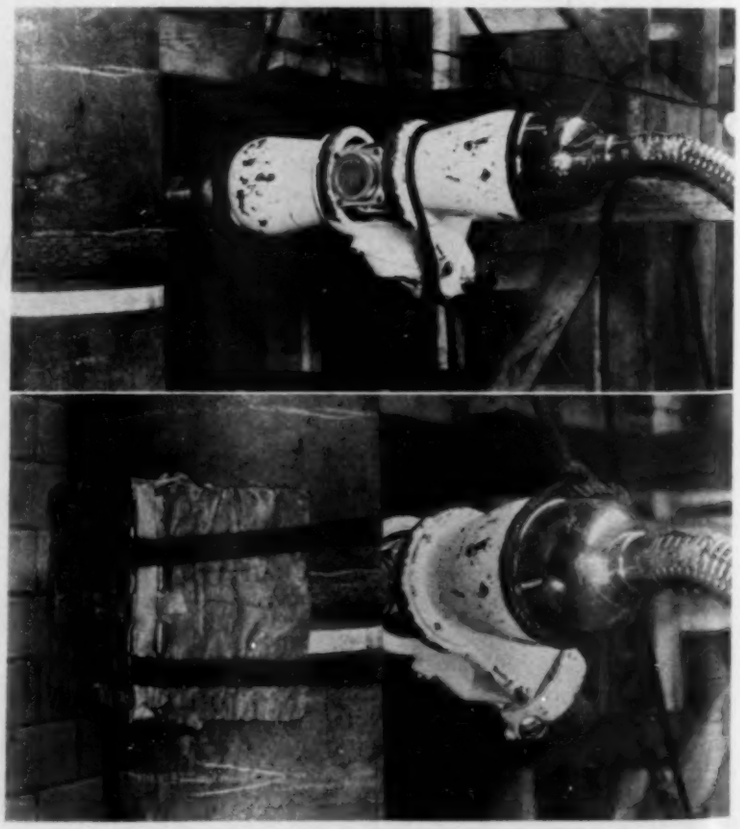


Fig. 9 (above)—Housed x-ray tube . . .
Fig. 10 (below)—Tube in position

The stress value given above is the total stress which would be obtained if no cold springing were used.

In the matter of piping support, convenience of erection and mechanical reasons prompted the adoption of constant support spring hangers as illustrated in Fig. 8.

With the usual type of spring supports employing helical or semi-elliptical springs, variations in hanger loading must be accepted as the unavoidable result of the rise and fall of the pipe during the change from cold to hot condition. The interchange of loading among the hangers is accomplished by bending in the pipe which in turn, depending on the direction of the motion, causes an increase or decrease of the pipe stresses. Inasmuch as the adopted layout already consumed the entire safe stress set as a basis for the design, it was felt that the hangers should offer a constant supporting effort to avoid any further increase in pipe stresses.

Aside from guides installed to protect the superheater headers, no anchorage was used as it was possible to keep the pipe thrusts at both terminals within the limitations established by the boiler and the turbine manufacturer.

The use of a special mixing chamber near the inlet of No. 4 turbine required the installation of a cross connection between both headers which, acting as a heating line will keep both lines simultaneously expanded in case of a shutdown of one of the two boilers of the unit.

X-Ray Examination of Welds

In order to assure us of a sound welded joint, free from cracks, gas pockets, slag inclusions, and other detrimental defects, some non-destructive method of weld examination had to be employed. X-ray examination proved to be, in our estimation for the time, a good reliable means for accomplishing this.

All joints on the most important lines, such as the main steam, the boilerfeed, and the 200 lb per sq in. bleed and station steam tie lines where welded were x-rayed, and where the exposures revealed faults, the metal was chipped out thoroughly, the hole repaired by welding and the entire joint stress relieved again. It was then x-rayed again and unless found to be sound, the repair procedure was repeated until the joint was proved sound. At first, several repairs were necessary, but as the work progressed the operators' technique improved greatly so that finally no faults were revealed. Consequently, the x-raying was discontinued a short time after the start of erection of the second unit.

The x-ray equipment used on this work was unique to the extent that it provided for an extremely high powered portable equipment and tube capable of great mobility, and especially adapted to piping work in the field under cramped conditions and close quarters.

The equipment containing the 220,000 volt transformer and water cooling apparatus was housed and set up on an automobile trailer chassis which could be hauled to the job, moved inside the building and placed in position near the work being done. Sometimes it was on the floor, or on a platform or scaffolding, and at times hoisted in the air and held by chain blocks. A special shockproof armored cable containing the leads

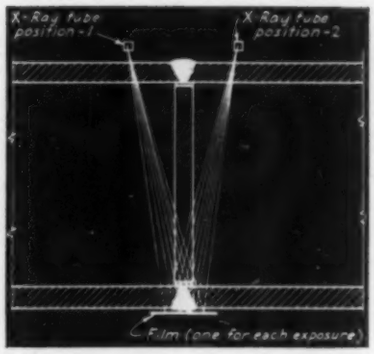


Fig. 11—Method of taking exposures for radiographic examination of welded joints

and concentric hose conveying the cooling water, extended for a length of 30 ft from this power source to the special shockproof metal housed high powered x-ray tube, Fig. 9. The tube was hung in place against the pipe at the welded joint to be examined, as shown by Fig. 10.

Rays from a point source within the tube passed through a window in the tube housing, through the pipe wall adjacent to the tube, and finally through the weld in the opposite wall, causing an exposure on a film held within a lead-covered film container strapped to the outside of the pipe. A sufficient number of exposures was taken all around the joint to include the entire joint, and shooting of the rays was done at an angle with two exposures taken for each examined section of the weld, as shown by Fig. 11. This procedure permitted depth of vision which enabled determining the location and depth of any fault showing up in the weld. Due to excessive wall thickness, it was necessary to examine the 20 in. welded joint at the turbine throttle by use of radium and the gamma rays emitted therefrom.

In Conclusion

The first unit has been operating successfully for about a year. The second unit installation has just been completed. As far as the piping is concerned the only conclusions drawn have been that welding of the low pressure lines (5 to 150 lb per sq in.) did not prove economical. We reverted to flanged construction of these lines on the second unit, and discontinued x-raying the welded joints. The indications thus far are that the design and equipment will prove to have been correct and that the desired long time satisfactory and economical operation will result.

Check Charts for Air Conditioning Calculations

By **J. M. Dean***

A GENERAL introduction to this series of charts appeared in the May, 1938, H. P. & A. C. and two of the charts have been published in each subsequent issue. Charts 7 and 8 are reproduced on the next page.

Chart 7 is used for checking sensible and latent heat gains due to ventilation air, and grains per lb to be removed from ventilation air. Chart 8 is for determining grains per lb of air entering the conditioned space and other psychrometric conditions of the air entering the space.

*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

[See preceding page]

Chart No. 7

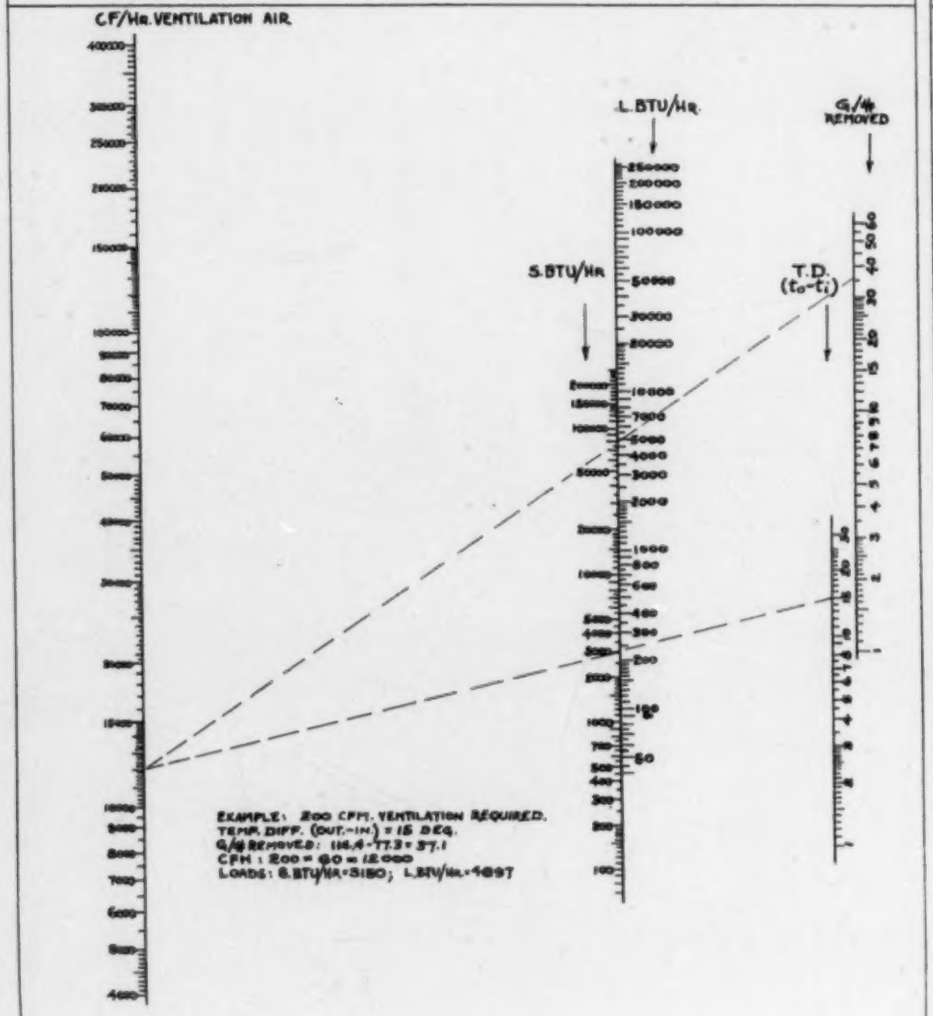

C.F/Hr. VENTILATION AIR
L. BTU/HR.
G/# REMOVED
S. BTU/HR
T.D. ($t_o - t_i$)
EXAMPLE: 200 CFM. VENTILATION REQUIRED.
TEMP. DIFF. (OUT.-IN.) = 15 DEG.
G/# REMOVED: 114.4 - 77.3 = 37.1
CFM = 200 × 60 = 12000
LOADS: S.BTU/HR = 5180; L.BTU/Hr. = 4097

Chart No. 8

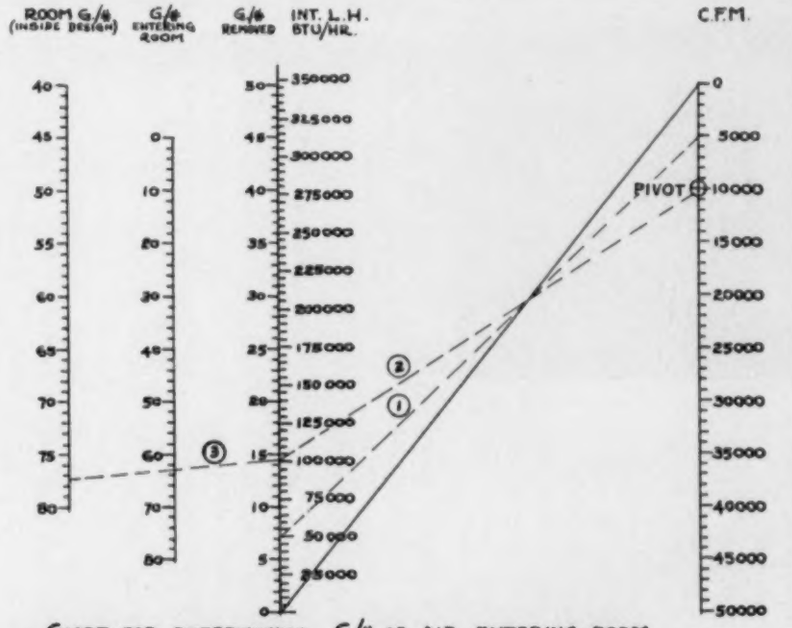


CHART FOR DETERMINING G/# OF AIR ENTERING ROOM.

EXAMPLE:
ASSUME ROOM G/# (INSIDE DESIGN) = 77.3; INT. L.H. = 50000; C.F.M. = 5000.
1. CONNECT 50000 INT. L.H. POINT & 5000 C.F.M. POINT.
2. DRAW LINE THRU PIVOT POINT & INTERSECTION OF LINE 1. WITH DIAGONAL; PROJECT THIS LINE TO "G/# REMOVED" SCALE.
3. CONNECT G/# REMOVED POINT & ROOM G/# POINT, 77.3 IN THIS CASE; AT INTERSECTION OF LINE 3. WITH "G/# ENT. ROOM" SCALE, READ 62.9.

NEXT DETERMINE D.B. TEMP. OF AIR ENTERING ROOM:
$D.B._{ENT.RM.}$ = D.B. ROOM (INSIDE DESIGN D.B.) − DIFFUSION TEMP. DIFFERENCE

USING ABOVE TWO DETERMINED VALUES (G/# AND D.B.) REFER TO PSYCHROMETRIC CHART FOR OTHER PROPERTIES OF AIR ENTERING ROOM.

INITIALLY DETERMINED VALUES		OTHER VALUES OBTAINED FROM PSYCHROMETRIC CHART					
D.B.	G/#	W.B. TEMP.	T.H., BTU/#	PERCENT R.H.	DEW POINT	G./CU.FT.	CU.FT./#

Specifying and Measuring Noise

By H. F. Hagen*

Examples of noise measurement in air conditioning and heating work are given, with information on the proper way of writing specifications for noise

EQUATIONS for sound intensity and decibels were given in August, with a discussion of the principles of noise measurement.[1] Sound meters and the three zero reference levels in use were also described. We continue this month with information on noise measurement and specifications in air conditioning work.

Assume that our room intensity, I_1, showed 40 db (decibels) to the A.S.A. reference zero, and that when our fan reached full speed the fan noise added to the room noise showed 45 db, which is our intensity I_2. We want to know now the decibels chargeable to the fan noise intensity, I_f. We know that (as is true in general)—

$$I_2 = I_f + I_1 \text{ and } I_f = I_2 - I_1$$

We also measured I_1 and I_2, so that—

$$40 \text{ db} = 10 \log \frac{I_1}{I_o}$$

$$4 = \log \frac{I_1}{I_o}$$

$$I_1 = 10000\ I_o = 1.0 \times 10^{-12} \text{ watts per sq cm}$$

Similarly—

$$45 \text{ db} = 10 \log \frac{I_2}{I_o}$$

$$4.5 = \log \frac{I_2}{I_o}$$

$$I_2 = 31600\ I_o = 3.16 \times 10^{-12} \text{ watts per sq cm}$$

Therefore—

$$I_f = 31600\ I_o - 10000\ I_o = 21600\ I_o,$$

and the fan noise to the A.S.A. zero is—

$$\text{db} = 10 \log \frac{21600\ I_o}{I_o} = 10 \log 21600 = 10 \times 4.34,$$

or, as we don't bother with fractions of a db,—

Fan db = 43 to A.S.A. zero;
= 36 to threshold of audibility;
= 29 to a zero of 1 millibar.

Position of Microphone

Even when we keep our zeros straight we still have another unstandardized factor in the position of the microphone with reference to the fan and the size and acoustic properties of the room in which the measurement is made. Sound intensities tend to vary inversely as the square of the distance from the source, and would do so exactly in a perfect medium in an open space with no reflection. The effect of the size and acoustic properties of the room can be minimized if the pick-up microphone is placed close enough to the fan.

At the present time, therefore, there is no such thing as the "noise of a fan in decibels." To have a meaning the statement must read somewhat more completely as, "The fan causes a noise indication of so many decibels above the A.S.A. zero with the pick-up microphone placed in such and such a location with reference to the fan inlet or outlet."

We see specifications with decibels as a factor providing that at a given location in the room to be ventilated the increase in decibels with fan running shall not exceed a specified number. This is not so bad. An increase of db is independent of the reference zero and the location is determined. But to make the statement complete the expected room level should be stated. For example, in the problem we worked above we had a room level of 40 db, a fan noise of 43 db and an increase of 5 db due to the fan. If the room level was 35 db, the room intensity would be—

$$35 = 10 \log \frac{I_1}{I_o}$$

$$3.5 = \log \frac{I_1}{I_o}$$

$$I_1 = 3160\ I_o$$

and with the same fan—

$$I_2 = I_f + I_1 = 21600\ I_o + 3160\ I_o = 24760\ I_o.$$

The noise in the room with the fan operating would then be—

$$\text{db} = 10 \log \frac{24760\ I_o}{I_o} = 10 \times 4.394 = 44$$

So, for an increase in room noise with a given fan noise of 43 db, we have 5 db if the room level is 40 db, and 9 db if the room level is 35.

(The increase in db for any two known intensities can be calculated directly from—

$$\text{db} = 10 \log \frac{I_2}{I_1}$$

In the last case where $I_2 = 24760\ I_o$, and $I_1 = 3160\ I_o$,

$$\text{db} = 10 \log \frac{I_2}{I_1} = 10 \log \frac{24760\ I_o}{3160\ I_o}$$

$$= 10 \log 7.83 = 10 \times 0.894 = 8.94$$

$$= \text{say 9 as before)}$$

Specification for Noise

The specification for noise should then be written as follows:

"The fan shall not increase the noise in the room by more than A decibels referred to a room level of B decibels. For these measurements the pick-up microphone shall be located one foot from the plane of the grille and center-lined therewith, and the diaphragm shall be placed in a vertical plane perpendicular to the plane of the grille and one foot from the extreme edge of the grille."

Such a statement would be completely defining. Just

*Vice-president in Charge of Research, B. F. Sturtevant Co.

[1] "Noise Measurement," by H. F. Hagen, HEATING, PIPING AND AIR CONDITIONING, August, 1938, pp. 533-534.

what values should be assigned to A is not known at present, but the values of B can be approximated from published information.

It must not be considered that, in the absence of standardization, noise measurement is of little value. On the contrary, the commercial value of noise measurement as a guide to design and installation is tremendous. Each investigator adopts methods which are standard in his work and serve design purposes as well as any other standards would. The important matter is an accurate instrument, which is readily procurable today.

Noise from Unit Heaters and Ventilators

A question is often asked in connection with units, such as unit ventilators or unit heaters. If two or more are placed in the room, how much will the additional units increase the noise in the room? The answer is—ten times the logarithm of the total number of units. If two units are used, in as much as the logarithm of two is 0.30103, the addition of the second unit will increase the noise 3 db; with three units, (log 0.477), the two additional units will give 5 db rise in noise level. Actually the rise will be somewhat less owing to the fact that the room could not be noiseless even without the units and therefore the noise intensity is not doubled by the addition of the second unit. A numerical example may make the point clear.

A school room with a noise level of 40 db is to be ventilated with two units. It is found that a figure of 45 db represents the effect of one unit. What will be the noise level in the room with one unit, and with two?

$$40 = 10 \log \frac{I_1}{I_0}$$

$$I_1 = 10000\ I_0$$

$$45 = 10 \log \frac{I_u}{I_0}$$

$$I_u = 31600\ I_0$$

$$\text{db} = 10 \log \frac{41600\ I_0}{I_0} \qquad (\log 41600 = 4.619)$$

db = 46, or a rise of 6 db with one unit.

With two units—

$$\text{db} = 10 \log \frac{I_1 + 2I_u}{I_0} = 10 \log \frac{(10000 + 2 \times 31600) I_0}{I_0}$$

$$\text{or db} = 10 \log \frac{73200\ I_0}{I_0} = 10 \times 4.864$$

say 49 db, or a rise of approximately 3 db for the addition of the second unit.

In sound calculations of the type that interest us, remember that it is the intensities that can be added or subtracted, never the decibels. Also, be sure to use the complete common logarithm which consists of the mantissa which you get from the tables and the characteristic, a whole number one less than the places to the left of the decimal point.

Example — log 3. = 0.477
log 30. = 1.477
log 300. = 2.477
log 3000 = 3.477
etc.

You have probably noticed the amounts of the intensities given for the zero levels and the watts per sq cm as used in one of our numerical examples. You may have been surprised at the almost infinitesimal amounts of energy involved. The energy in even loud sounds is extremely small. The velocity of propagation is high, 1100 ft or 34,000 cm per second, but the oscillatory or wave pressures and velocities are minute. A classical illustration of the smallness of the vibration instances a beam of sunlight in which the motes are visible. The passing of the wave of even a loud sound through this beam does not disturb the motes an amount sufficient to be visible.

Useful Data

The following data you may find useful:

Density of air under usual conditions of temperature and humidity in grams per cc:

$\rho = 0.0012$

Velocity of sound in cm per second:

$C = 34000$

ρC as in Equation 1 on page 533 of the August issue = 40.8

Noise in db (A.S.A. zero) in various locations:[2]

Boiler Factory	105-115
Average Factory	50- 90
City Street	50- 81
City Office	40- 70
City Residence	30- 55
Country Residence	30- 40

[2]From "Physics Forum," "Noise Measurement," by Dr. E. E. Free.

Omaha U Has Air Conditioning

The new Municipal University of Omaha Building at Omaha, Nebraska, is "one of the finest examples of automatic year 'round air conditioning, including summer cooling and winter heating" according to the "Business Conditioner" issued by the Nebraska Power Co. A description of the installation appears in the August number of the publication.

The summer cooling cycle uses well water from two deep wells on the campus for refrigeration, augmented when necessary by one 40 ton and one 50 ton compressor. Each well supplies 120 gpm of 50 deg water to a chilled reservoir from which the water is pumped to a cooler situated in the equipment room, and thence to cooling coils in each of the two conditioners. From the cooling coils it passes through the condensers of the refrigerating compressors. The water, now warmed to about 80 deg, is then discharged into the lawn sprinkling reservoir.

The air supply of the air conditioning system is provided by two main supply fans capable of delivering 110,000 cfm of air, of which one-fourth may be outside air if desired. Separate pneumatic temperature controls are provided for each room to assure accurate regulation for the requirements of student classes and when the room is vacant.

Winter heating and conditioning is taken care of through the same system of air distribution as that for summer cooling except that heating coils supplant the cooling coils. Humidification is provided by automatically controlled steam jet humidifiers. The same automatic control system for summer cooling is used for heat and humidification control in winter.

Temperature Regulation in Modern Fruit Storage

By Clarence E. Baker*

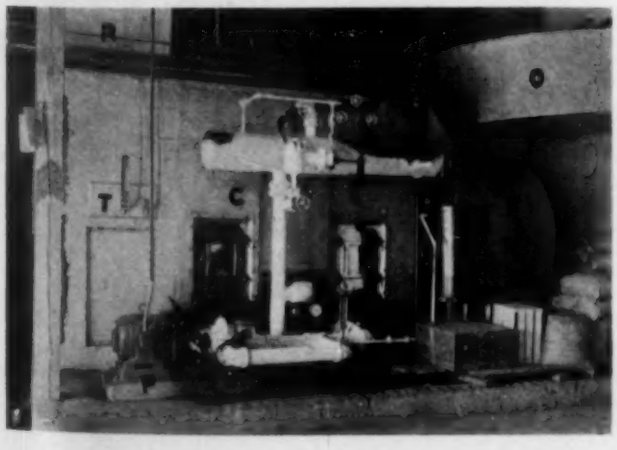
Modern cold storage plants making use of air conditioning principles solve the problem of maintaining low temperatures and high relative humidities in a satisfactory manner when they are properly designed and efficiently operated. It is not always realized, however, that comparable conditions can be obtained through different control methods which may or may not be equally efficient when all factors are considered; too frequently, a job is operated in one way year after year with little regard to the factors that increase or decrease operating expenses. . . . The author has had the opportunity to make some studies of the various means of securing thermostatic temperature regulation in a modern fruit storage and he presents a summary of results here

THE refrigeration of fruit and vegetable products requiring low temperature and high relative humidity often presents a difficult problem. Modern cold storage plants, making use of air conditioning principles, are solving this problem in a satisfactory manner when properly designed and efficiently operated.

In the case of apple storage the temperature range of 31 to 35 F at a relative humidity of 85 to 90 per cent often can be nicely obtained by wet coil cooling units, or what is sometimes called the "brine spray air blast system." Such systems are composed either of small air conditioning cabinets, usually placed in the refrigerated room, using one or more to the room depending upon the size of the room and the capacity of the unit, or a central air washer of large capacity. In the latter arrangement the air is cooled and conditioned in the central unit and carried to the rooms to be refrigerated by means of insulated ducts. The cold air absorbs heat from the storage rooms and is returned to the air washer where it is reconditioned and recirculated to the rooms. With the individual room units the air is circulated within the room and through the units for reconditioning.

The cooling units, whether large or small, contain the low pressure side of the refrigeration system. The refrigerant evaporates in direct expansion piping or one of several other forms of evaporators. Over these cold surfaces brine is sprayed, cooled and recirculated within the unit by means of a brine pump, which forces it under pressure through a system of spray nozzles. The air used to cool the storage rooms is passed through the chilled brine spray. The brine prevents ice formation on the evaporator surfaces, eliminating the necessity of defrosting encountered in the case of dry coil units.

Fig. 1—The central air washer used in the cold storage described. (R) Return air duct from the storage rooms. (O) Discharge duct from air washer carrying cold air to the storage rooms. (F) Housing of ventilating fan that forces the air through the system. (C) Side of air washer containing the direct expansion piping and brine spray. (P) Brine pump. (T) Thermostats



The term "air blast" is rather misleading as it implies a high velocity movement. In practice in apple storages the air movement is maintained at as low a velocity as possible, except early in the season when a large amount of warm fruit is being stored and the field heat must be extracted quickly. High velocity air movement, especially if the relative humidity is low, tends to cause shriveling of the fruit if continued for long periods.

In the automatic operation of brine spray systems of this type the room temperature may be regulated by placing the active element of the thermostats in various locations. In the case of room units the thermostat ordinarily is operated by the room temperature over the differential for which the thermostat is set. Each unit generally has its own thermal valve so that the units may be operated individually or in groups depending upon the installation. Where a large central air washer is used as the source of refrigeration for various rooms the problem becomes more complicated, especially if rooms of different temperatures are to be maintained in the same system. This may be accomplished within limits by varying the amount of air admitted to different rooms by a suitable damper arrangement.

The temperature of the air in the distributing system may be controlled thermostatically in various ways. The temperature of the evaporating refrigerant, the temperature of the brine in the air washer, the temperature of the air leaving the air washer or that of the air returning to it may serve as the basis of controlling the air in the storage rooms with varying conditions and with varying degrees of convenience, accuracy and efficiency.

An opportunity to investigate several of these factors in a 55,000 bushel apple storage, composed of three rooms operated from a central air washer, presented itself as a phase of another line of work. Refrigeration for this

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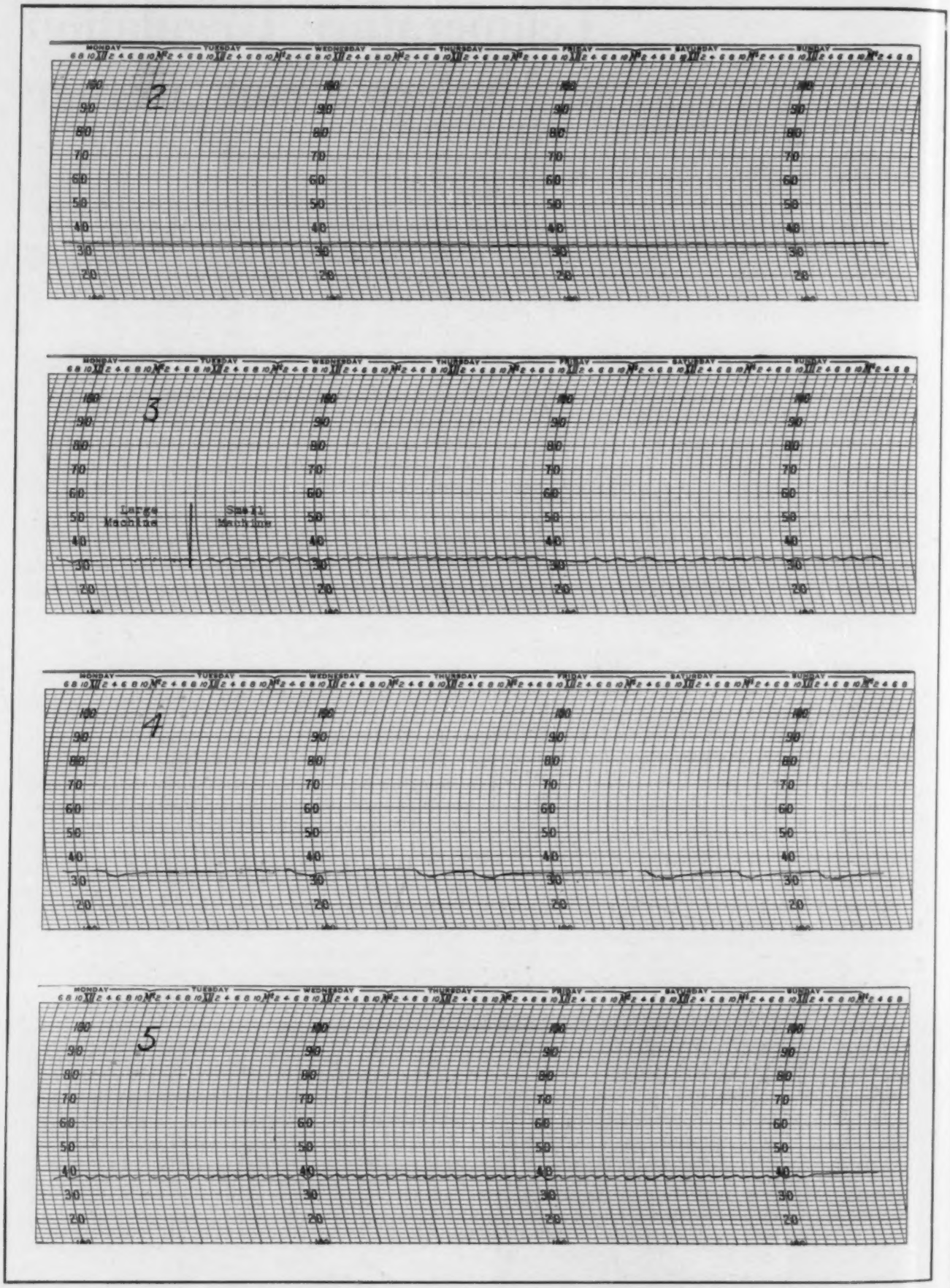

Figs. 2-5—Temperature charts for the various arrangements investigated

plant is furnished by one 6½ x 6½ in. ammonia compressor with a 30 hp motor and one 5 x 5 in. auxiliary compressor with a 15 hp motor. The air is circulated through the air conditioner shown in Fig. 1 and into the duct system by means of a two speed ventilating fan with a capacity of 28,000 cfm at high speed at 10 hp and about half that amount at low speed at 5 hp. Low speed is used except in emergencies where more rapid cooling is desired. One room is operated at 36-40 F and the other two at 32-34 F under normal conditions.

During the experimental set-ups described below the temperature recording instruments were placed along the center aisle just within the fruit stack out of the main line of air movement about 5 ft above the floor. Consequently, they were influenced by the temperature of the fruit surrounding them as well as that of the air circulating through the storage rooms.

Each compressor is operated by a thermostat, the activating bulb of which is in the brine below the evaporating coil inside the air washer. A third thermostat operates the brine pump, controlling the brine spray as occasioned by the location of its activating bulb.

Effect of Location of Thermostat Bulb

Under the first experimental arrangement the bulb of the brine pump thermostat was situated in the air return duct where it was exposed to the temperature of the air returning from the storage rooms. This arrangement was not considered satisfactory for the operating conditions desired as a variation of 3 to 4 deg in room temperature resulted between cycles of refrigeration. This likely could have been overcome but another arrangement was considered more practical.

The second arrangement simply consisted of moving the brine thermostat bulb to the discharge air duct between the brine spray and the fan. When the temperature of the air being discharged from the washer reached the high setting of the thermostat instead of starting the brine pump the thermostat started the compressors by means of a relay. Refrigeration was produced in the expansion coils of the air washer, cooling the brine until the discharging air reached the temperature corresponding to the low point of the thermostat setting.

The brine pump operated continuously, maintaining the brine spray at all time. Because of the large volume of air in the system, totaling approximately 152,000 cu ft, the room temperature could be controlled very accurately by this method. After the refrigeration load had become fairly constant the temperature of the air could be carried very near to that desired in the 32-34 deg room, usually about 29 to 30 deg.

This uniform temperature is illustrated by the chart in Fig. 2 over the period of one week, during which very little fluctuation occurred. The buffering effect of the large volume of air is very great. This may be illustrated by the fact that on several occasions more than 3000 bushels of apples at temperatures of 75 to 90 F were placed in one room during a period of only a few hours without any detectable variation in the air temperature of the room or at any point in the system. The large quantity of cold brine constantly circulating through the spray system within the washer provided a second buffering effect, serving to prevent rapid changes in air temperature. The air going through the ventilating system passed through the cold brine spray whether or not refrigeration was being produced in the expansion coils.

The ventilating fan also was operated constantly and aided in maintaining a uniform temperature in all parts of the room and avoided the formation of air pockets.

The compressors were operated simultaneously or one at a time by means of a combination manual and automatic arrangement. Individual automatic compressor operation could be secured by setting the thermostats for different ranges so as to cut on and off at different temperatures.

The third arrangement involved cutting the brine pump and the compressor on and off simultaneously without changing the location of the thermostat bulbs. When the air in the discharge duct from the air washer reached the low setting of the thermostat the compressor was shut down and the brine spray system as well. By this practice the buffer effect of the cold brine was lost and the air temperature rose rapidly between refrigeration cycles. To maintain a uniform temperature in the storage rooms the large compressor was operated with a narrow differential setting of the thermostat, as illustrated at the beginning of the temperature curve in Fig. 3. This involved starting a 30 hp compressor motor and a 7½ hp brine pump motor several times an hour. To avoid this the thermostat was set for a slightly wider differential and the small compressor put into service and permitted to operate over a longer period. Under these conditions the compressor would operate for about an hour and a half and be idle for about the same time. This avoided some of the undesirable effects of the extremely short and frequent operation cycles and gave a temperature curve that was satisfactory from a practical standpoint where accurate temperature regulation is essential. The temperature chart for these operating conditions is shown in Fig. 3 from Tuesday noon on.

Under the fourth set-up the plant was operated by a room thermostat set for a variation of 2-4 F room temperature. This thermostat was placed near the center of the storage room on a post surrounded by stacked fruit where it would respond to the air dry bulb temperature prevailing in that location. The purpose of this arrangement was to reduce the frequency of operation cycles as far as possible with the minimum sacrifice of temperature uniformity. The room thermostat controlled the fan operation. When the fan motor was shut off a relay from fan to brine pump motor cut off the brine and a relay from brine motor to compressor motors shut off the compressors. The chart in Fig. 4 illustrates a typical week of operation during the period when there was a small or negligible live load. Ordinary heat transfer or leakage accounted for most of the temperature rise as the fruit had been held at storage temperatures for more than four months.

When the compressor was placed in operation the temperature of the room would be reduced to the low setting of the thermostat in from two to five hours and the temperature would rise to the high setting of the thermostat in from eight to 36 hours, depending upon conditions. During these periods of rising temperature, however, it will be noticed that the initial rise to near the upper setting of the thermostat took place relatively quickly. The curve then flattened out near the higher end of the range and often continued in this fashion for an extended period before the thermostat operated to start the compressor. In other words, during

the long periods between operating cycles the temperature of the room was close to the high point of the temperature differential for a large portion of the time, thus making the mean temperature prevailing near the high point of the range. In accurate temperature control under this manner of operation this fact would have to be considered. It was necessary to carry a lower air temperature by this intermittent system of air circulation to hold the fruit at a given temperature in comparison with an arrangement using a constant movement of air. This made relative humidity regulation more difficult, as it is easier to maintain the high relative humidity desired with a narrow differential between room temperature and that of the incoming air. Conditions that vary from day to day also influence the frequency of the operating cycles under this form of operation, as is shown by the chart. (Fig. 4).

Fig. 6—A portion of one of the storage rooms. The cold air is discharged from the metal duct at the ceiling over the center aisle toward the sides of the room. The air is removed from the room through a duct beneath the slatted floor of the center aisle

The type of temperature regulation illustrated in Fig. 5 was secured in another apple storage using two individual wet coil units in the refrigerated room. The thermostat was governed by the room temperature on a 2 F differential and, as the chart shows, the temperature was held in a very uniform manner. Under these conditions the compressor operated from 30 to 45 minutes out of approximately each 3 hr period.

Results of Different Operating Methods

From the standpoint of the effect upon the fruit any of the charts shown represent very satisfactory conditions, with the possible exception of that shown in Fig. 4. This would be considered a satisfactory chart if the condition desired was a 34 F temperature permitting a negative fluctuation. Minor temperature fluctuations of short duration have little effect upon the temperature of the stored fruit. For all practical purposes the extreme accuracy shown in Fig. 2 is not essential and the condition represented in Fig. 3 is equally desirable for commercial work. Remote registering recording instruments with the bulb placed well back into the fruit stack do not respond to air temperature variations in the order of those shown in Figs. 3 and 5, but record a uniform temperature, usually 1 to 3 F above the mean air temperature after the field heat has been well removed.

Engineer Must Decide

The engineer must determine what system of plant operation is most efficient for his conditions. Frequent starting of large motors with a high starting load may not be as efficient a practice as longer periods of operation spaced further apart, or the continuous operation of brine pumps and fans with small motors. These observations and charts are presented not as concrete examples of all the factors involved but to show that, many times, comparable conditions can be obtained through different methods which may or may not be equally efficient when all the factors are considered. Too frequently a job is operated in one way year after year with little regard to the factors that increase or reduce operating expenses.

[These observations were made in the cold storage plant of the Johnson Orchards, Mooresville, Ind., through the courtesy of the owners.]

Conditioning Units Cool Chapel

Wherever people work, live or gather indoors there is a fertile field for modern air conditioning. In fact, air conditioning has proved itself as valuable in a house of worship as anywhere else. This is demonstrated by the chapel of St. Louis University where portable air conditioners have been installed.

Six of the eight conditioning units in chapel at St. Louis University

Eight units are installed along the wall to regulate temperature and humidity. The chapel is 50 by 65 ft with a seating capacity of 200.

Air Conditioning and the Income Tax

By Lawrence R. Bloomenthal*

Methods of computing depreciation rates of air conditioning equipment are reviewed, and some of the fundamentals of income tax matters as they affect air conditioning are discussed for owners and installers of systems

AIR conditioning is undergoing such rapid development that it is difficult to predict with absolute accuracy the length of time that present installations will retain their usefulness. Despite the complexities involved, however, owners must have an estimate of the probable useful life of the system. This is essential as a basis for working out the annual rate of depreciation for income tax purposes.

Computing Depreciation

Under present laws, a taxpayer is permitted to deduct annually a reasonable amount from his gross income to compensate him for the gradual deterioration of property used in his business. In this way, a reserve can be built up so that the original investment will be recovered by the time the property wears out. Ordinarily, a reserve for depreciation is set up as an account on the books and does not represent an actual sum of cash set aside for making replacements. While building up a cash reserve might be the best assurance that new machinery would be installed as needed, most businessmen prefer to use their cash as working capital and finance replacements in other ways. The rate at which the annual deductions are made depends upon the estimated useful life of the improvement.

Straight Line Method—The plan most widely used for calculating depreciation is to divide the original cost of the property by the number of years of its life. This is known as the "straight line" method because it assumes that an equal amount of wear and tear takes place annually. The rule is simple to apply when it involves but a single piece of machinery: a $10,000 pump, for example, may have a life of 20 years so that 1/20 of the original cost can be taken each year. This means that the annual rate for depreciation is 5 per cent of the total cost, or $500.

Composite Rate—It is customary practice to submit to the prospective purchaser of air conditioning a lump sum estimate covering labor, machinery, materials and so forth. This not only relieves the bidder of explanations as to the comparative costs of various units, but also avoids the necessity of submitting lengthy and detailed schedules.

When the property owner sets up this expenditure on his books, only the lump sum figure for the entire job is shown. This simplifies the preparation of the depreciation schedule which must be attached to the state and federal income tax return. What is known as a "composite rate" will be calculated by dividing the total cost over the number of years it is estimated the system will continue to provide efficient service. By doing this, the entire installation is treated as if it were a completely integrated unit, such as a single motor or pump.

This method of computation has certain disadvantages. For one thing, a composite rate may result in either more or less depreciation being taken than might be allowed if each of the individual parts were being depreciated separately. Suppose a 20 year life is assigned to an air conditioning system. The sheet metal ducts and the remodeling necessitated by their installation may cost $20,000. This work probably should be considered an addition to the building itself and ought to be depreciated over its remaining life, which might be 30 years. Accordingly, 1/30 of the $20,000 should be taken annually from gross income as depreciation. But, when a composite rate for the system is employed and the client does not have an itemized estimate of costs, there is no way of knowing what proportion of the total price was paid for the ducts. Instead, they are depreciated over the 20 year life of the system, thereby taking more depreciation than would otherwise be allowed. The government has been allowing such deductions but it may eventually decide against the use of rates working out this way, and taxpayers will be forced to revise their computations and may be subjected to additional tax liability.

There is also the risk of taking less depreciation under the composite rate than will be adequate to provide for the actual wear and tear that is taking place. A blower fan, for instance, might normally last 20 years if given ordinary care. But, when this same fan is incorporated into a particular air conditioning system, its useful life may be shortened considerably and it may, for the purpose of making our point, hold up only five years. If the composite rate is based on a 20 year life, and the fan lasts only five years, it is obvious that the property owner will not have allowed for sufficient depreciation on that item.

Deducting Losses—Another drawback in considering the system as a unit and dividing its cost over the number of years of its estimated useful life arises when an entire piece of machinery wears out before it was expected to and must be replaced. Since the taxpayer has no record of individual costs, there is no accurate basis for taking advantage of the provisions in the income tax laws allowing a loss to be claimed for business property which loses its useful value and must be discarded. In order to claim the benefit of this loss, it is necessary to acquire information as to the original cost. After a lapse of five or six years, it may be quite difficult for the firm which installed the system to produce the desired figures upon request. If it is not possible to segregate the cost of the discarded machine,

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no loss can be deducted, but the taxpayer can continue to take depreciation on the original basis just as if that piece of machinery were still in service.

Unit Method—Of course, if the cost of every single item of machinery were to be set out, there would be a great deal of extra bookkeeping required. For example, a typical installation in a large office building may require several miles of piping and sheet metal ducts as well as a large number of motors and other moving parts. According to the "unit method" of depreciation, each of these items would have to be recorded separately on the depreciation schedule and a separate computation made for the annual allowance for wear and tear. Under this system, every fan, motor and piece of equipment must be listed and depreciated individually.

While the unit method probably will give the most accurate picture of the depreciation, it is clumsy to use. Not only must the engineering firm compile cost specifications, but the property owner also would have to go through the cumbersome procedure of making detailed calculations to compute the total deduction. This method would involve the least additional work for smaller buildings where installations are not so complicated.

Depreciation Rates—Depreciation rates for various items of equipment are set out in the table. It must be remembered, however, that these rates may be increased or decreased depending upon the circumstances in each particular case. Additional depreciation is allowed for unusual wear and tear.

A Practical Plan—The third and probably the most practical plan of computing depreciation is to see that a classification is made of the different types of machinery with all those of a similar nature being placed in the same class. According to this arrangement, if there were 10 high speed motors of approximately the same capacity, all of them would be grouped together and their cost would be segregated from the total amount paid for the installation. If these motors were worth $7500, then this sum would be divided by the average life of all the motors in the group which might be about 15 years. The rate would be 1/15 or 6⅔ per cent of $7500, which would permit $500 annually to be taken as depreciation. The same procedure would be followed for each of the other classes.

Depreciation Rates for Various Items of Equipment[a]

Type of Equipment	Annual Depreciation Rate, Per Cent
Air Conditioning system in hotels or restaurants (composite rate)	6⅔
Blower or exhaust fans	6⅔ to 10
Pumps (varies with types)	4 to 6⅔
Compressors	5
Cold storage plant (composite rate)	6⅔
Cooling towers	6⅔
Electrical equipment:	
Conduits, wiring and fittings	4
Motor controllers	8
Motor (varies with capacity):	
Large units, over 3000 kva	3½
Medium units—1000 to 3000 kva	4
From 50 hp to 1000 kva:	
High speed	5 to 6
Low speed	4
Below 50 hp capacity	6 to 7
Meters (recording type)	6⅔
Power feed wiring	10
Switchboards and wiring	5
Magnesia insulation on hot pipes	10
Water purifier	10

[a]Abnormal usage may justify increasing these rates to allow for as much as 15 to 30 per cent annual depreciation. The table is based on actual decisions of the courts and Board of Tax Appeals, and a bulletin issued by the Bureau of Internal Revenue outlining rates which can be used in the average situation. It should be clearly understood, however, that no fixed and arbitrary rates can be stated which will apply to every motor, compressor or fan. Depreciation varies with the circumstances of each particular installation of machinery and the rates fluctuate accordingly.

Service Contracts

Once an air conditioning system has been installed and is in operation, the firm which handled the job often enters into a service contract with the owner. According to this agreement, the owner pays a fixed fee in return for which inspections of the equipment are made at regular intervals. Some property owners are under the misapprehension that these payments form part of the cost of the system and must be capitalized. This is not true since any sums paid out under this contract are part of the ordinary maintenance expenses and may be deducted from gross income in the same manner as the cost of all ordinary and necessary repairs.

In connection with service contracts, the question arises whether repair work ordered at the suggestion of the inspector constitutes an "ordinary and necessary repair" or is actually an addition or replacement. Repairs, of course, are deductible in full in the year they are made while additions or replacements become part of the capital asset and must be depreciated over the remaining life of the entire system. So long as the only effect of the work done is to keep the machinery operating efficiently and to prevent excessive deterioration due to neglect, a deduction for repairs can be claimed. But, if obsolete or worn out parts are replaced so that either the useful life of the equipment or the value of the system is materially increased, the law considers this to be a "capital improvement." In other words, it is not so much the cost of the work, but the extent and character of the changes that are made which will determine whether it is a repair or a capital improvement.

Repair Expense

An interesting point sometimes raised by prospective purchasers is whether any part of the cost of installing an air conditioning system can be charged off to building repair expense. This is especially true if the building is in a somewhat run-down condition so that the installation of this equipment will rehabilitate the property and prolong its life expectancy. There have been a few cases in which this same contention was raised and the courts have ruled that if there was any basis for segregating the repairs from the capital improvements, a deduction in full could be taken for the former and only the balance would be considered a capital improvement whose cost would have to be recovered by annual depreciation.

The rulings on this point have been quite general, but they involve a definite principle which can be applied to air conditioning installations. Whenever a property owner can show that a certain portion of the cost expended for the system really went toward repair work not directly connected with its operation or with the installation itself, that amount can be deducted in full from gross income as a current expense.

Suppose, then, that an installation is made in a 20 year old building which because of its age also needs considerable repair work. A single firm takes care of the entire job and bills the owners for $70,000. If it can be proved that $10,000 went for necessary repairs

[Concluded on p. 598]

What Weather Does to Man

By William F. Petersen, M. D.*

CARVED over the fireplace of the banqueting room of the old Rathaus of Lübeck is this inscription—"Many a man sings loudly when they bring him the bride. Knew he what was being brought, he might well weep."

Certainly no Jeremiah would I be, but occasionally I do harbor the notion that in a few respects air conditioning as now decked out in all its wedding habiliment, may be such a bride. Her dower may be most attractive and life with her may be more comfortable, but whether the descendants of the marriage will be so able might be questioned!

What Does "Weather" Do to the Normal Individual?

It might be well to make inquiry what actually happens to the normal individual when he is not air conditioned—when our spoiled child of the 20th century is removed from the cocktail bar or the movie and exposed to "nature in the raw" in the way of ordinary atmospheric changes in the northern storm tracks.[1] This promptly becomes one of the most complex of questions for the very simple reason that the extremes of environmental variability (polar as contrasted to tropical air-mass effects) may in some instances bring about *similar* biological reactions, while, on the other hand, like environmental changes (for instance, the passage of repeated polar air masses), may bring about diametrically *opposite* biological reactions in the inadequate individual.

Within certain limits the passage of a polar air mass "stimulates" the normal individual. That is, there is first a period of "retreat" to the interior of the body, the skin vessels contract, the blood pressure increases, the vascularization of the interior is greater, the blood becomes transiently more alkaline, the K/Ca ratio increases, the body cells become less permeable. Then, because certain tissues have not been getting enough oxygen in regions where the blood vessels have been partly closed, and because of the interplay of endocrine activity, the entire organism reverses its metabolic trend, and the tissues reveal *stimulation*, associated with dilation of the vessels; the blood then becomes more acid, the pulse and respiratory rate and the basal metabolic rate are increased, the K/Ca ratio is lowered. In other words, a biologic phase is set up which underlies the feeling of stimulation and the increased "drive" of the northern regions which Huntington has so well documented.

For the biologically inadequate individual, the energy expenditure involved by the passage of the atmospheric front may mean the onset of disease. (By this I mean the awareness of inadequacy; the actual pathological disturbance may have existed previously). So, too, the very young individual and the old individual will be more susceptible and of course the sicker the sick individual, the greater the susceptibility. So-called weather-sensitive individuals (and that usually involves the more highly differentiated types, the very slender or the very fat, *i.e.*, the extreme types) are those who have a very labile (*i.e.*, "liable to lapse") autonomic (self-governing) nervous system, which, when subjected to disturbance, reacts in a variety of ways, usually annoying, but not necessarily fatal (migraine, arthritis, asthma, colitis, urticaria, a change in the mood or disposition, etc.).

The "Tropical Effect"

By contrast to the effect of the cold air masses which I have merely sketched, the effect of the tropical air mass naturally dilates the peripheral vessels, lowers the blood pressure level, relaxes smooth muscle, lowers the basal metabolic rate. If prolonged, or the temperature is unusually high, a relative alkalosis may follow, with greater irritability of the autonomic and central nervous system. The studies of Sundstroem and Mills are interesting in this connection.

On the whole, the summer (tropical) status is one that is associated with increased well being. During the summer and autumn we reach our highest peak of resistance, as revealed in the depression of the mortality curve. Our reserve of buffers is greatest, our vitamine accumulation highest.

But during this time, because of the increase in autonomic irritability,[2] an unusual environmental disturbance, the effect of cold, for instance, is followed by much more profound effects in some in-

As interest in the medical and health aspects of air conditioning grows, opinions differ widely, for (as Dr. Petersen points out here) the human being is the most individualistic of animals. Some see in air conditioning the possibility of preventing many illnesses and the ultimate prolongation of human life; others are concerned with the effect of such factors on the human system as the shock when entering or leaving a space where the temperature is maintained much lower than the outside. Some studies have indicated noteworthy reductions in sickness with air conditioned offices; similar studies in other buildings have failed to do so. . . . There is no question, however, that the subject is one that increases in importance every day. To give those concerned with air conditioning a background of information on the effect of weather on man, Dr. Petersen hits some of the high spots in this brief discussion. The author is a keen student of—and has written extensively on—the effect of the weather on the patient

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[1] I take it for granted that the reader is fully conversant with the details of the cyclonic circulation of the atmosphere and the implication of environmental variability in the storm track region; it is the only region with which I am concerned in this discussion.

[2] Probably due to higher bioelectrical potential differences of cell membranes, with the possibility that when biochemical break occurs, the disturbance is more profound.

dividuals than the actual change in the environment would apparently warrant.

This particular fact becomes of importance because the only real insult offered the population by the air conditioning engineer is the vast differential that exists at certain times between outside temperature and the designed indoor air temperature; under present practice this may vary by as much as 30 F. Physiologically this is preposterous. While within the range of adaptation for the perfectly normal individual, it involves a definite metabolic strain and can be harmful for a considerable group of subjects.

The vasomotor crisis associated with the rapid adjustment necessary under such conditions may (a) cause an increased susceptibility to infection, (b) result in gastro-intestinal disturbance, (c) precipitate acute dysfunction in organs that are inadequate.

The reversal from a sojourn in an air conditioned (cool) environment of this type to the hot, humid, outside environment will in turn *accentuate* the peripheral dilatation that will normally follow a period of unusual and unseasonal chilling—with resulting excessively low blood pressure, fainting, nausea, etc.

Nevertheless the engineer is quite justified in assuming that changing air mass as now designed has no great significance for the normal individual and for this group can fabricate the air conditioning structure either for greater comfort or greater efficiency, or both, but he should keep in mind that even here unusually abrupt environmental changes may be of pathological significance under certain conditions.

Differences in Sex and Human Types

It should be remembered by the engineer that one of the reasons that the human became a human was because he lost his hairy coat. That forced the skin to become very active in shunting the blood back and forth between skin and brain and the large viscera; brought about greater vasomotor activity; developed the necessary autonomic mechanisms and in so doing gave the opportunity for the brain to become just as active as the environment was unstable. Our ape ancestor who lost his hair got sense when he got cold! (I use the male gender quite properly because on the whole the normal female[2] is less susceptible to environmental change than the male; this is due largely to her greater fat padding, *i. e.*, her insulation is better. This in turn is biologically proper because it makes her metabolism more stable, thus insuring more equitable conditions for the development of the offspring. It is this biological difference that permits the female to discard clothing to the extent that she does, the while her brother clings to his heavies and his overcoat in the winter time.)

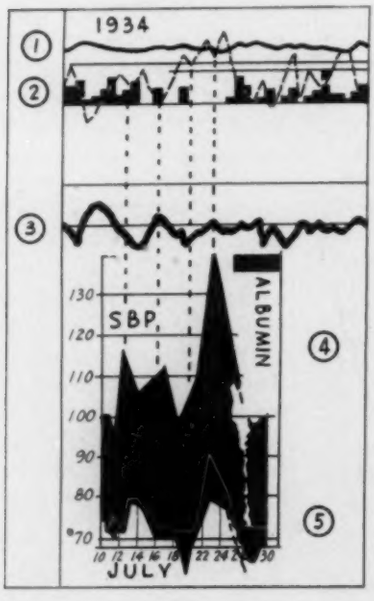

Fig. 1—Chart of day by day observations of blood pressure revealing the fact that even during hot weather the blood pressure may increase; this might occur when a patient of this type were subjected to air conditioned rooms with a differential of 20 to 30 F. The chart shows the blood pressure curve of a hospitalized patient (not air conditioned) during the period of the unusual heat wave in Chicago, 1934. Note blood pressure increase during this period (actual rise occurs as a result of the seemingly minor polar episodes indicated by the dotted vertical lines leading from the crests of the blood pressure curve to the depressions in the temperature curve above) . . . In the graph the upper trace is that of the wind velocity (1), the dotted line the daily mean temperature (2), the heavy curve the daily barometric trace (3), the systolic and diastolic blood pressure is indicated by the black field (4), the pulse rate by the white line in the black field (5). Note that with the infall of cold air following the period of unusual heat, the patient develops albuminuria

[2]The slenderized female does not come into this category.

Apart from the difference inherent in sex, differences in the habitus of individuals must be considered. It might seem self-evident that the extremely slender individual is more susceptible to cold and will become fatigued more readily with repeated effects; that the fat individual is more susceptible to heat because he cannot radiate as well—but again let me stress that this individual also cannot stand the improperly air conditioned room in summer, for if suddenly exposed to this chilling effect, his blood pressure rise may be so great that dire cardiac and vascular damage may result.

In touching on this subject I must briefly discuss another physiological phase, namely, the fact that the "comfort zone" will depend on the vasomotor tone of the individual and that the individual with "cold feet" has a general contraction of his skin vessels and feels cold "all over," no matter what the air temperature. We see this exaggeration in the individual suffering from a chill, as for instance, from malaria. Here the peripheral vessels are in spasm, the blood supply to the skin is markedly impaired, and the individual feels cold even if his internal temperature is 106 to 107 F.

The man with cold feet may feel cold with a room temperature of 80 F! The obvious remedy is both simple as well as complex. We can at least see to it that the air flow in the region of the legs is not too rapid and that the temperature of the air mass between the ceiling and the floor level is equalized.

We can provide a certain amount of radiant heat in the scheme of heating and not depend wholly on general air temperature for control. If the individual of the type that we have here under consideration (and this usually involves the older members of the family and the slender types) normally wants some place to "toast his feet" on the day when the barometric pressure is high, there would appear to be sound physiological reason for this desire and the engineer should take this into consideration. Usually the warm glow of the electric heater is adequate and practical. Much more difficult is the effective and proper method of actually heating the floor. I have recently had the opportunity of observing some of the excavations made at Trier and at Cologne and noted the care with which panel and floor heating was provided for the wealthy Roman

[Concluded on p. 598]

Engineer Describes Method He Uses for Filing Technical Articles and Papers

Many engineers have developed methods of filing magazine articles, technical papers and other data to meet their personal needs. William Goodman* describes here the system he has employed for several years

THERE are several ways of arranging a filing system for technical papers and magazine articles. For this reason, a description of one that has met my needs for several years may also be of interest to others. No claim is made that the system described here has been planned logically or that it will meet the needs of everyone. It represents a haphazard growth that has answered my own peculiar requirements and may be useful to others.

A number of folders labelled according to the items in the "List of Folders" are kept in a filing cabinet. A separate folder is provided for each item in the list, and the subject and its number are marked on each folder. The folders are numbered for two reasons: (1) By

*The Trane Co. Member of Board of Consulting and Contributing Editors.

List of Folders

A.
- 10 Acoustics
- 20 Air, Properties of
- Air Conditioning
  - 30 Air Distribution
  - 40 Chemical
  - 50 Controls
  - 60 Costs
  - 70 Drying
  - 80 Duct Work
  - 90 Fans
  - 100 Filters
  - 110 Heat Gains, Body
  - 120 Heat Gains, Miscellaneous
  - 130 Heat Gains, Solar
  - 140 Humidification
  - 150 Ice
  - 160 Industrial
  - 170 Installations
  - 180 Instruments & Measurements
  - 190 Maintenance & Operation
  - 200 Miscellaneous
  - 210 Noise
  - 220 Psychrometric Charts & Data
  - 230 Residence Research
  - 240 Spray Chambers & Air Washers
  - 250 Standards
  - 260 Systems & Layouts
  - 270 Theory & Calculations
  - 280 Trouble Shooting
  - 290 Ventilation

B.
- 10 Bearings
- 20 Bibliography
- Boilers
  - 30 Coal Tests
  - 40 Hartford & Return Connections
  - 50 Miscellaneous
  - 60 Oil & Gas Tests
  - 70 Selection & Sizing

C.
- 10 Charts
- 20 Chimneys
- 30 Coal
- 40 Combustion
- 50 Corrosion

D.
- 10 Drawing Standards, Etc.

E.
- Electric & Gas Contracts
  - 10 Standard Forms
- 20 Electrical
- 30 Elevators

F.
- 10 Filters
- 20 Flow of Fluids

G.
- 10 Gages

H.
- Heat Transfer
  - 10 Building Construction
  - 20 Coils
  - 30 Data
  - 40 Mathematics
- Heating
  - 50 Central Plants
  - 60 Controls
  - 70 Cost and Economics
  - 80 Exhaust Steam
  - 90 Hot Water
  - 100 Instructions
  - 110 Intermittent
  - 120 Meters
  - 130 Miscellaneous
  - 140 Piping
  - 150 Radiation Calculations
  - 160 Radiator Orifices
  - 170 Steam Traps
  - 180 Trouble Shooting
  - 190 Unit Heaters
  - 200 Vacuum Pumps
  - 210 Warm Air

L.
- 10 Lubrication

M.
- 10 Machine Design

O.
- 10 Oil Burners

P.
- 10 Plumbing
- Power Plants
  - 20 Condensers
  - 30 Diesel Engines
  - 40 Miscellaneous
  - 50 Piping
  - 60 Steam Engines
  - 70 Turbines
- 80 Pumps, Centrifugal

R.
- 10 Reducing Valves and Pressure Regulators
- Refrigeration
  - 20 Absorption Plants
  - 30 Applications
  - 40 Codes and Standards
  - 50 Compressors, Reciprocating
  - 60 Compressors, Centrifugal
  - 70 Condensers, Evaporative
  - 80 Condensers, General
  - 90 Cooling Towers and Spray Ponds
  - 100 Controls and Expansion Valves
  - 110 Evaporators
  - 120 Kelvin Reverse Cycle
  - 130 Lubrication
  - 140 Miscellaneous
  - 150 Piping
  - 160 Refrigerants
  - 170 Refrigerants, Freon
  - 180 Steam Jet
  - 190 Thermodynamics
  - 200 Trouble Shooting

S.
- 10 Stokers

T.
- 10 Temperature—Measurements and Instruments
- 20 Thermodynamics

U.
- 10 U. S. Weather Bureau Reports

V.
- 10 Vacuum Cleaners—Central System

W.
- 10 Water Treatment
- 20 Wells

BUILDINGS
- 10 Auditoriums and Exhibition Buildings
- 20 Bakeries
- 30 Banks
- 40 Breweries
- 50 Churches and Memorials
- 60 Factories
- 70 Garages
- 80 Hospitals
- 90 Hotel and Apartment Buildings
- 100 Laundries
- 110 Lodge Buildings
- 120 Motion Picture Studios
- 130 Office Buildings
- 140 Printing Plants
- 150 Schools
- 160 Theaters
- 170 Transportation

marking each magazine article or paper with the number of the folder in which it belongs, filing mistakes are avoided. (2) Those who have assistants need only mark the folder number on an article of interest and then pass it along to others for cutting and filing. By marking each article from the list, which should be kept handy, a stenographer can attend to all the details—cutting, filing, bringing folders when needed, and returning articles to the file.

A magazine article that is to be kept is cut from the magazine, stapled in the left hand corner, and placed in the proper folder. While this system requires some work to start, it requires the least work and bother for its maintenance. Folders containing magazine material can easily be removed and replaced. Furthermore, adding new folders is a simple matter.

All of the folders are filed alphabetically. As soon as a new folder is added, a number is assigned to it. If a new folder is to be filed under, say, "M" between two folders already numbered "10" and "20," the new folder is marked "M-11," or with any other number between 10 and 20. In the case of subheads, as under "Air Conditioning," the main heading "Air Conditioning" appears on each folder along with the subheading; there are no folders for the main heading alone. Under the heading "Buildings" at the bottom of the list, descriptive material relating to actual buildings is filed. All of these "Building" folders are filed together behind the letter "Z."

The size of the system is limited only by the size of the filing cabinet. For a small system, an inexpensive, heavy paper transfer case provides as much filing space as one file drawer.

Weather—

[Concluded from p. 596]

when he lived in the north woods of Europe in the second and third centuries. I am firmly convinced that the comfort that can be achieved (mental and physical) when the individual can live in a room with relative warm floors and cooler air, will in the final analysis prove really advantageous.[4]

Health and Air Conditioning

Finally, a few words concerning "health and air conditioning." Too often there is much loose discussion concerning the possible reduction of "illness" for individuals working in buildings that are air conditioned; by illness there is usually implied a reduction in colds and respiratory infections in general.

There is at present no good evidence, in my opinion, to support such broad statements. While the theoretical possibility exists that continued and absolute uniformity of optimal environmental conditions should render the individual more resistant (though even this thesis could be attacked) such a situation never exists in real life. At best the artificially created situation exists for eight hours a day and at least eight more are spent under conditions where the environment varies greatly. The opportunity for crowd infection[5] is many times as great on the street car and subway, in the restaurant, the shops, and the movies and the home, as it normally would be in the office or factory. A reduction in morbidity need not be anticipated merely because we air condition the office or factory for one-half of the individual's ordinary daily period of action.

With ordinary methods of heating (with the usual high and dry temperature of the room), the mucous membranes may be dehydrated and resistance to bacterial penetration may diminish; it would then seem logical that with proper air conditioning the mucous membranes would be kept in a more normal condition and resistance to infection would be increased.

Actually, however, an infection involves not only the presence of the proper virus on the mucous membranes, but is preceded by a period of generalized as well as localized vascular spasm before the bacteria apparently penetrate. Such periods of vascular spasm are by no means controlled (though they may be modified) by merely placing the individual in an air conditioned room and we must not expect prevention of all colds and respiratory infections by such means. In other words, our enthusiasm should be moderated by the realization that we seldom deal with simple situations, but very complex relations indeed.

The discussion of this important and interesting field would require too much space and I can only suggest to the engineer that the human organism happens to be an amazingly intricate and at times seemingly unreasonable piece of apparatus and that he must not expect it to behave according to the rules of the handbook of mechanics. The course of a million years cannot be changed over night even by the most ardent air conditioner. The more reasonable engineer will learn to be patient with the most individualistic of animals—his human subject.

[4]The English, have of course, developed panel heating but have done little as yet in the way of actual floor heating.

[5]Assuming that infection is the only factor involved in the onset of illness, which it is not.

Income Tax—

[Concluded from p. 594]

to the building itself, that amount probably will be allowed as a deduction. This situation should not be confused with the tearing out of otherwise good walls or partitions, re-arranging electrical conduits and incurring other expenses which would be unnecessary except for the installation. All of these things are incidental to the main improvement and their cost is part of the cost of the system.

H.P.&A.C. Piping Review

THE accompanying tabulation supplements similar data given on page 108 of the February, 1938, HEATING, PIPING AND AIR CONDITIONING. The object is to place pertinent information on piping installations on the record for convenient and general reference.

It is hoped to publish similar information on other installations as it becomes available. Suggestions for making these summaries of the greatest usefulness will be appreciated by the editors.

Summary of Information on Piping Installations

NAME AND LOCATION — ENGINEERS AND CONTRACTORS	PIPING SERVICE	PRESSURE AND TEMP. LB PER SQ IN. AND F	NOMINAL DIAM. AND WALL THICKNESS IN.	COMPOSITION OF PIPE MATERIAL	FLANGED ELECTRIC OR GAS WELDED?	STRESS RELIEVED?	JOINTS X-RAYED?	KIND OF GASKETS
Essex Station, Newark, N. J.	High pressure steam	1500 lb, 1000 F	Sched. 160	Carbon-molybdenum	Electric welded	Yes	Yes, for jts. 6 in. and over	1/16 in. annealed iron
Piping Contractors, United Engineers & Constructors. Piping Fabricator, The M. W. Kellogg Co.	Int. main steam and low pressure boiler feed	300 lb, 750 F	10 in. and under. Sched. 40; 18 in., 24 in. and 26 in.—1/2 in., 30 in.—5/16 in.	Carbon steel	Electric welded	Yes	Yes, for jts. 6 in. and over	1/16 in. Durabla
	Boiler feed pump	75 lb, 750 F	10 in. and smaller. Sched. 40; 12 in. and larger Sched. 30	Carbon steel	Electric welded	No	No	1/16 in. Durabla
	Exhaust steam	15 lb, 450 F	10 in. and smaller Sched. 40; 12 in. to 20 in. inc. Sched. 30; 24 in.—5/8 in.	Carbon steel	Electric welded	No	No	1/16 in. Durabla
	High pressure boiler feed	2000 lb, 450 F	Sched. 160	Carbon steel	Electric welded	Yes, on jts. 5 in. and over	Yes, on jts. 6 in. and over	1/16 in. annealed iron
	High pressure blow down	1500 lb, 600F	Sched. 160	Carbon steel	Electric welded	Yes, on jts. 5 in and over	Yes, on jts. 6 in. and over	1/16 in. annealed iron
	Blow down	900 lb, 600 F	Sched. 80	Carbon steel	Electric welded	Yes, on jts. 8 in. and over	No	1/16 in. annealed iron
	Safety valve blow off	50 lb, 950 F	10 in. and smaller Sched. 40; 12 in. and over Sched. 30; 24 in. and 36 in.—5/8 in.	Carbon steel	Electric welded	No	No	1/16 in. Durabla
Waterside Station No. 2, Units 4 and 5, New York City.	High pressure steam	1400 lb, 900 F	Sched. 160	Carbon-molybdenum	Electric welded	Yes	Yes	None
Piping Contractors, M. H. Treadwell Co. Piping Fabricator, The M. W. Kellogg Co.	Boiler feed discharge	1725 lb, 600 F	3/4 in. to 6 in. Sched. 160; 8 in., 10 in. and 12 in. Sched. 140	Carbon steel	Electric welded	Yes	Yes	None
	Boiler blow off	900 lb, 750 F	Sched. 80	Carbon steel	Electric welded	Yes	Yes	Kell-Raph
	High pressure bleed and steam	250 lb, 750 F	Sched. 40	Carbon steel	Electric welded	Yes	Yes	Kell-Raph
	Inter bleed or fresh water	125 lb, 450 F	12 in. and smaller Sched. 40; 14 in., 16 in. and 18 in. Sched. 30	Carbon steel	Electric welded	No	No	1/16 in. Consolco
	Low pressure bleed exhaust	125 lb, 400 F	12 in. and smaller Sched. 40; 14 in., 16 in. and 18 in. Sched. 30	Carbon steel	Electric welded	No	No	1/16 in. Consolco
Schuykill Station, Philadelphia, Pa.	Main steam	1350 lb, 925 F	Sched. 160	Carbon-molybdenum	Electric welded	Yes	Yes	Annealed Armco iron
Consulting Engineers, United Engineers & Constructors. Piping Contractors, M. W. Kellogg Co.	Boiler feed	1700 lb, 400 F	Sched. 160	Carbon steel	Electric welded	Yes	Yes	Annealed Armco iron
	Cross-over	225 lb, 500 F	Various up to 30 in.	Carbon steel Carbon-molybdenum	Flanged and welded	Yes	Yes	
American Smelting & Refg. Co., Maurer, N. J.	Main steam	140 lb, 750 F	Extra heavy	Carbon steel	Flanged	No	No	Kell-Raph
Piping Contractors, M. W. Kellogg Co.	Boiler feed	520 lb, 350 F	Extra heavy	Carbon steel	Flanged	No	No	1/16 in. Durabla
	150 lb steam	150 lb, 425 F	Std.	Carbon steel	Flanged	No	No	1/16 in. Durabla
	Boiler blow off		Extra heavy	Carbon steel	Flanged	No	No	1/16 in. Durabla
	5 lb steam	5 lb, 220 F	Std.	Carbon steel	Flanged	No	No	1/16 in. Durabla
	Fuel oil	250 lb, 180 F	Extra heavy	Carbon steel	Flanged	No	No	1/16 in. Durabla
	Safety valve discharge		Std.	Carbon steel	Flanged	No	No	1/16 in. Durabla

[Continued on next page]

Name and Location Engineers and Contractors	Piping Service	Pressure and Temp. Lb per Sq In. and F	Nominal Diam. and Wall Thickness In.	Composition of Pipe Material	Flanged Electric or Gas Welded?	Stress Relieved?	Joints X-Rayed?	Kind of Gaskets
Monongahela West Penn. Public Service Co., Rivesville, W. Va.	High pressure steam	1350 lb, 935 F	3 in.—0.437 in. 6 in.—0.794 in. 10 in.—1.25 in. 12 in.—1.147 in.	Carbon-molybdenum	Welded joints	Yes	Yes	
Consulting Engineers, Sanderson & Porter. Piping Contractors, M. W. Kellogg Co.	High pressure boiler feed	1700 lb, 450 F	Sched. 160	Carbon steel	Flanged and welded	Yes	No	Annealed iron
	Turbine exhaust	400 lb, 750 F	20 in. x 1/2 in. 20 in. x 3/4 in.	Carbon steel	Flanged and welded	No	No	Durabla
	Low pressure steam Feed exh.	350 lb, 250 lb, 125 lb	Std.	Carbon steel	Flanged and welded	No	No	Durabla
Chile Exploration Co., Tocopilla, Chuquicamata, Chile.	Boiler feed pump suction		Std.	Carbon steel	Flanged	No	No	
Piping Contractors, M. W. Kellogg Co.	Boiler feed discharge	900 lb, 379 F	Extra heavy	Carbon steel	Flanged	Yes	No	
	Turbine exhaust	205 lb, 635 F	Std.	Carbon steel	Flanged	Yes	No	
	Gland steam	205 lb, 635 F	Std.	Carbon steel	Flanged			
	Auxiliary steam	205 lb, 635 F	Std.	Carbon steel	Flanged			
	Emergency supply		Std.	Carbon steel Wrought iron	Flanged			
	Drain and vent		Std.	Carbon steel Wrought iron	Flanged			
	Stage heating		Std.	Carbon steel	Flanged			
	Blow off		Sched. 80	Carbon steel	Flanged			
	Soot blower	205 lb, 635 F	Sched. 40	Carbon steel	Flanged			
	Fuel oil		Std.	Carbon steel	Flanged			
	Safety valve discharge		Spiral welded pipe	Carbon steel	Flanged			
	High pressure drains	750 lb, 875 F	Sched. 80	Carbon steel Carbon-molybdenum	Flanged			
	Compressed air		Sched. 40	Carbon steel	Flanged			
West Virginia Pulp & Paper Co., Charleston, S. C.	Superheated steam	600 lb, 725 F	3 in. to 16 in.—0.300 in. to 0.843 in.	Grade A A106-36	All three	Yes	No	Copper
	Boiler feed	730 lb, 220 F	3 in. to 8 in.—0.216 in. to 0.322 in.	Grade A A106-36	Gas and electric welded	Yes	No	Copper
	Saturated steam	150 lb, 370 F	2 in. to 18 in.—0.154 in. to 0.375 in.	Grade A A106-36	Gas and electric welded	Yes	No	Comp. asbestos
	Saturated steam	35 lb, 280 F	2 in. to 24 in.—0.154 in. to 0.437 in.	Grade A A106-36	Gas and electric welded	Yes	No	Comp. asbestos
	Auxiliary steam	600 lb, 725 F	3 in. to 6 in.—0.300 in. to 0.432 in.	Grade A A106-36	Gas and electric welded	Yes	No	Copper
	Condensate	15 lb, 220 F	1/2 in. to 10 in.—0.109 in. to 0.307 in.	Grade A A106-36	Gas and electric welded	Yes	No	Copper

Essex Station

All systems were shop fabricated as far as possible.

While entire job is marked "electric welded," there were some flanges used; these were of the welding neck or slip on type.

On high pressure steam, carbon molybdenum forgings were used for branch outlets. On high pressure boiler feed, carbon steel forgings were used for branch outlets.

All welded joints made with chill rings.

Corrugated sections used for cross-over connections.

Carbon steel pipe and carbon moly pipe diameter and wall thickness per A.S.A. Std. B36.10-1935.

Carbon moly pipe made to A.S.T.M. Spec. A-158-36, Grade P-1. Carbon steel pipe made to A.S.T.M. Spec. A-106-36, Grade A.

A good deal of corrugated pipe was used for the intermediate main and low pressure steam systems.

Waterside Station No. 2

While entire job is marked "electric welded," there were some flanges used; these were of the welding neck or slip on type.

The first four systems listed were shop fabricated as far as possible. The last two systems listed were field fabricated as far as possible.

Carbon steel pipe and carbon moly pipe diameter and wall thickness per A.S.A. Std. B36.10-1935.

Carbon steel pipe made to A.S.T.M. Spec. A-106-36, Grade A and Grade B (depending on service conditions). Carbon moly pipe made to A.S.T.M. Spec. A-158-36, Grade P-1.

A good deal of corrugated pipe was used in the low pressure bleed and exhaust.

Schuylkill Station

This job was shop x-rayed but not shop hydro-tested and not field x-rayed but field hydro-tested.

While job is indicated as electric welded, there were some flanges used for the boiler feed lines. These were of the welding neck type.

Entire job was shop fabricated as far as possible.

All welded joints made with chill rings.

On high pressure steam, carbon moly forgings were used for branch outlets. On high pressure boiler feed carbon steel forgings were used for branch outlets.

Carbon steel pipe made to A.S.T.M. Spec. A-106-36, Grade B. Carbon moly pipe made to A.S.T.M. Spec. A-158-36, Grade P-1.

A good deal of corrugated pipe was used for the cross-over system.

American Smelting & Refg. Co.

Entire job was shop fabricated as far as possible.

Pipe made to A.S.T.M. Spec. A-106-36, Grade A or B.

Monongahela West Penn Public Service Co.

All welded joints stress relieved, both in shop and in field. All headers full furnace stress relieved. All material shop hydro-tested as well as field hydro-tested after erection. All accessible shop joints x-rayed

High pressure steam—carbon moly forged tees used for branch outlets. High pressure boiler feed—carbon moly cast tees and "headers" used for branch outlets.

All welded joints made with chill rings. Joints upset at ends for high pressure steam and high pressure boiler feed. Corrugated sections used for turbine exhaust and low pressure turbine leads.

Carbon steel pipe—Std. and Sched. wall thickness refer to A.S.A. B26.10-1935. All carbon steel pipe Grade B A.S.T.M. Spec. A-106-34T.

Carbon moly pipe—Special thickness of wall—Made to A.S.T.M. Spec. A-158-35T, Grade P-1.

Chile Exploration Co.

The first three systems listed were shop fabricated and the balance field fabricated.

Pipe made to A.S.T.M. Spec. A-106-36, Grade A and B.

Heating · Piping and Air Conditioning

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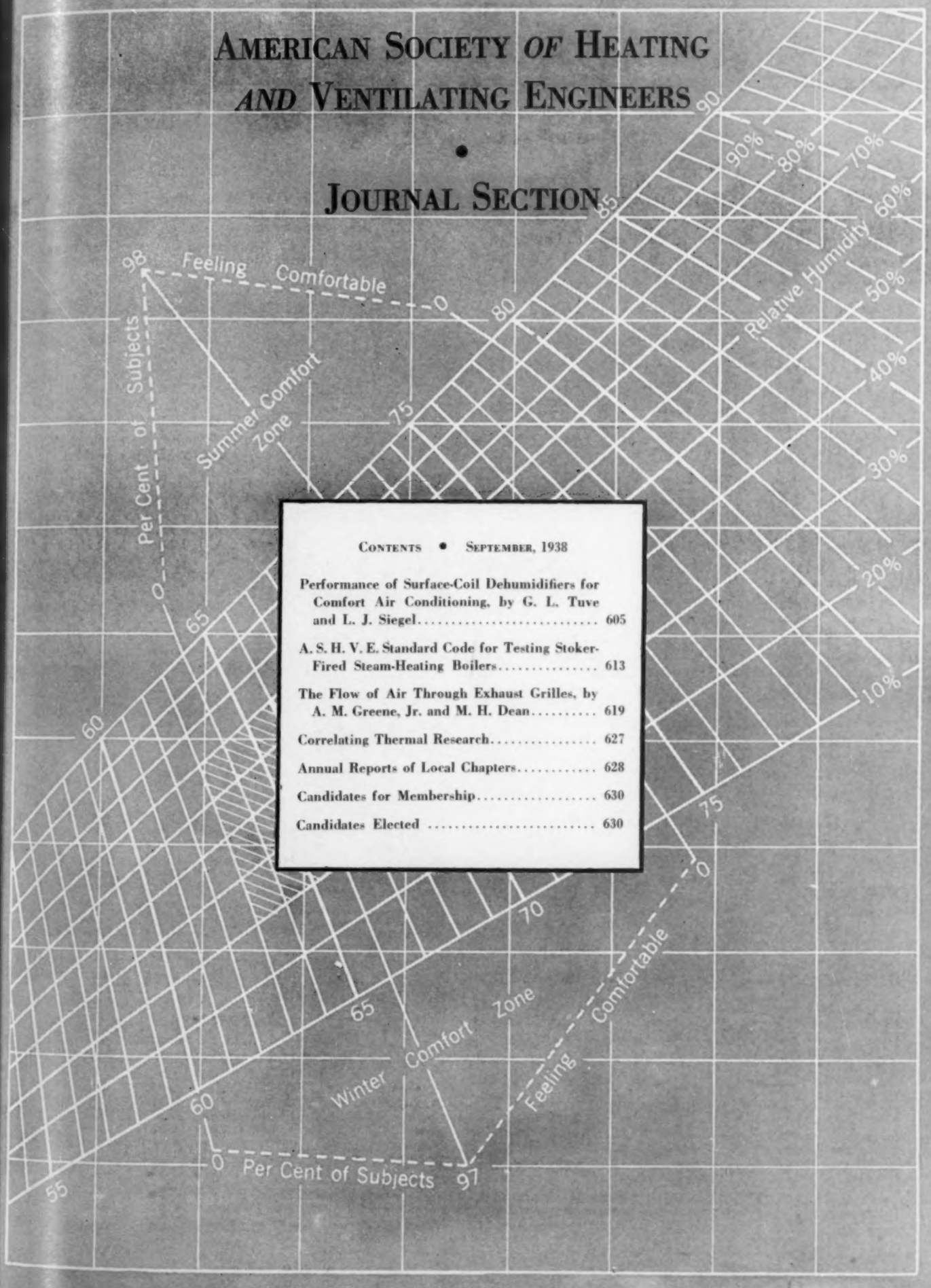
Journal Section



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Performance of Surface-Coil Dehumidifiers for Comfort Air Conditioning

By G. L. Tuve* (*MEMBER*) and L. J. Seigel,** Cleveland, Ohio

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Case School of Applied Science.

Introduction and Summary

EVER since extended surface coils were first introduced in air conditioning, the engineer has been confronted with the problem of finding some method for rating them and for calculating their performance after the ratings have been made. Because of the immediate need for a solution to the problem, a great number of methods have been devised and are now in use. These procedures are based on heat transfer theory to a certain extent, but most of them resort to many approximations, either because the theory does not apply, or because it becomes too complicated for practical use. The purpose of this paper is to review some of the methods available, and to propose another method which combines both simplicity and accuracy. This has been named the *humidity method*.

In general, the problem may be attacked in two ways: (1) Calculations may be based directly on overall coefficients of heat transfer, U or K. (2) Ratings and calculations may be made on the basis of individual surface coefficients on the refrigerant side and the air side, or on the basis of surface temperatures. If overall coefficients are used, the ratings are usually prepared in tabular or curve form, and special devices are resorted to for the calculation of dehumidification problems. When surface coefficients are used, the coil can be rated over its entire operating range by means of only a few tests, but the calculations for coil performance are usually more complicated because the methods using surface coefficients are based more exactly on heat transfer theory.

In the first report on this cooperative research project,[1] the two methods of analysis were examined, and the basic theory of dry-coil performance was reviewed. Test results available in the literature were summarized. In the second report,[2] additional experimental data on dry coils were presented, and a method was suggested as a basis for dry-coil rating and calculation, using overall coefficients. The third report[3] dealt with dehumidification, and showed the approximate variation of performance in the range of commercial coil designs. Here again the possibilities of several methods based on overall coefficients were examined. The present paper deals with the application of the surface coefficient method for dehumidifying coils. A simple method of solution on the psychrometric chart is given, and its applications to coil rating and coil selection are discussed. Solution by trial-and-error becomes unnecessary with this *humidity method*, and coil performance over the entire range can be determined from the results of a few simple dry-coil tests. The method is applicable to both direct-expansion and water coils.

One of the inherent advantages of surface-coil dehumidifiers over the spray type is the higher exit dry-bulb which is possible with a given amount of dehumidification. This may result in a smaller required size of refrigerating machine, and the necessity for little or no reheating. As soon as the performance characteristics of surface coils are better understood by air conditioning engineers, these advantages will be more widely recognized.

Correlations Between Dry-Coil and Wet-Coil Performance

A practical summary for comparing most of the methods now in use for figuring dehumidifying coils is given in Table 1. A single type of problem is used as a basis for this comparison, *i.e.*, the problem of finding the exit air conditions, the coil size and the required temperature of refrigerant when the inlet air conditions and the heat load ratio are known. In methods 1 to 4, in Table 1, the overall coefficient of heat transfer (U or K) is used. Methods 6 and 7 utilize the individual air-side and refrigerant-side coefficients. Methods 5 to 7 also employ some form of surface temperature in the calculation.

Reference is made in Table 1 to the advantages and disadvantages of the various methods, as indicated by the list in Table 2. An ideal method for rating and calculating dehumidifying coils would have all of the favor-

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[1]Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, by G. L. Tuve and C. A. McKeeman (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 427).

[2]Performance of Fin-Tube Units for Air Heating, Cooling and Dehumidifying, by G. L. Tuve (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 99).

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

[3]Performance of Fin-Tube Units for Air Cooling and Dehumidifying, by G. L. Tuve and C. A. McKeeman (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June 1937, p. 379).

Table 1—Seven Methods for Figuring Dehumidifying Coils

Type of Problem: ***Given,*** **inlet air conditions, total load, heat-load ratio, air velocity, refrigerant velocity, depth of coil.**

Required, **face area, exit air conditions, refrigerant temperature.**

Method No.	Test Results Used as Basis for Analysis	Procedure for Solution (See Equations, Table 3)	Advantages (See Table 2)	Disadvantages (See Table 2)	References (See Footnotes)
1	Overall coefficients for dry coil as a function of: (1) Air velocity; (2) Depth of coil; (3) Refrigerant velocity.	Assume refrigerant temperature. Solve for exit dry-bulb and for face area by Eqs. A and B, using sensible heat load only. Test answer by experience, and make other trials if necessary.	4, 5, 8	1, 2, 3, 6, 8, 9, 11	A.S.H.V.E. Guide 1937, p. 193
2	Overall coefficients as in No. 1. Wet-coil multipliers as a function of temperature differences.	Assume refrigerant temperature. Solve for exit dry-bulb and for face area by Eqs. A and B, using wet-coil coefficient and total heat load. Make other trials if necessary.	4, 8	1, 2, 5 7, 8, 9 10	Commercial bulletins
3	Overall coefficients as in No. 1. Wet-coil multipliers as a function of heat load ratio. Exit relative humidities as a function of heat load ratio.	Locate exit relative humidity on load ratio line. This fixes exit conditions. Solve for face area by Eq. B and for refrigerant temperature by Eq. A, using wet-coil coefficient and total load.	1, 3, 4, 6, 8	2, 5, 7, 8, 10	(3)
4	Overall coefficients for sensible and latent heat transfer, given separately. (Latent heat coefficients based on dew-point mean temperature difference.)	Assume refrigerant temperature. Solve for exit dry-bulb and exit dew-point by Eq. A, using U_s and U_L. If exit condition does not fall on load ratio line, make other trials.	3, 4, 5, 6, 8	1, 2, 7, 8, 10	(3)
5	Ratio of heat removed to heat removable, as a function of air velocity and depth of coil. Refrigerant - to - surface temperature differential as a function of total load, ($\triangle T$).	Using intersection of load ratio line with saturation curve as surface temperature, find exit conditions by ratio of heat removed to heat removable. Solve Eq. B for face area. Find refrigerant temperature by subtracting $\triangle T$ from surface temperature.	1, 3, 4, 6, 8	4, 5, 6, 8, 10	(6), (7)
6	Surface coefficients as a function of air and refrigerant velocities.	Assume refrigerant temperature. Find extent of dry surface, if any. Solve for final wet-bulb, and locate intersection on load ratio line. (Tables of empirical factors required).	1, 2, 3	4, 5, 8, 10	(4)
7	Surface coefficients as a function of air and refrigerant velocities.	Find minimum dry-bulb for zero dehumidification by Eq. F, with T_s = entering dew-point. Draw a relative humidity line through this dry-bulb and entering dew-point. Find face area from Eq. B, surface temperature from Eq. F and refrigerant temperature from Eq. D.	1, 2, 3, 4, 5, 6, 7, 8	8, 10	This Paper

able characteristics listed in the first part of Table 2 and none of the disadvantages enumerated in the last part of that table. From Table 1 it is evident that, while the early methods of figuring surface dehumidification had many advantages, they also had certain important disadvantages, chief of which were their lack of general application and their uncertain accuracy.

The proposed *humidity method* of coil rating and calculation (No. 7, in Table 1), has most of the advantages and few of the disadvantages listed in Table 2. This method deserves further investigation, as outlined in the last section following.

The procedure outlined in Table 1 for each of the seven methods is not in all cases the only procedure available. For instance, in method No. 2, an assumed wet-bulb depression at the exit, or relative humidity of the exit air may be used as the starting point, rather than an assumed refrigerant temperature. Again, in using method No. 5, the refrigerant-to-surface temperature differential is sometimes found by an arbitrary rule or approximation, rather than from tests.

There are also several types of problems other than the one used as a basis for Table 1 and the procedure for solving some of these is illustrated in Table 4. Trial-and-error solutions are necessary for many types of problems when methods 1 to 6 are used, while method No. 7 usually affords a direct solution, without trial-and-error.

Table 2—Advantages and Disadvantages of Various Methods of Figuring Dehumidifying Coils

(See Table 1)

Advantages (Characteristics of the ideal method)

1. Trial-and-error solution not required.
2. Few tests necessary for rating a coil.
3. Good accuracy for coils of the type tested.
4. Method easily explained from psychrometric chart and simple heat transfer equations.
5. Tables of empirical factors not necessary.
6. Easy graphical solution on psychrometric chart.
7. Wet-coil tests not required for establishing ratings.
8. Calculations very simple.

Disadvantages (of actual methods)

1. Many problems require trial-and-error solutions.
2. Large number of tests necessary for rating coils.
3. Uncertain accuracy, probably not closer than 10 per cent.
4. Method difficult to understand in terms of fundamentals.
5. Extensive tables of constants or empirical factors required.
6. Inaccurate for high latent heat loads.
7. Difficult tests required for establishing ratings.
8. Applicable only to specific coils tested.
9. Applicable only to operating conditions used in tests.
10. More research necessary to establish accuracy.
11. Coil may not dehumidify according to specified heat-load ratio.

Surface Temperatures and Surface Coefficients

Many of the solutions to surface-coil problems now in use have been developed through a knowledge of some form of surface temperature. The true surface temperature of a coil is a variable, depending on many factors in the coil's construction and operation. Goodman has shown that surface temperature varies in the direction of the air flow,[4] but equally important are the variations due to the temperature rise of the refrigerant and to the temperature gradient in the fins. Some investigators have attempted to determine surface temperatures by direct measurement with thermocouples or by theoretical calculation from other measured quantities.[5] To take the place of true surface temperature, fictitious surface temperatures have been worked out and used by different experimenters. For example, the intersection of the heat-load ratio line with the saturation line on the psychrometric chart has been given the interpretation of surface temperature by Knaus,[6] Wells,[7] Pownall[8] and

[4]Dehumidification of Air with Coils, by William Goodman (*Refrigerating Engineering*, October, 1936, p. 225).

[5]Extended Surface Cooling Units, by R. H. Swart (*Refrigerating Engineering*, February, 1938, p. 107).

[6]Heat Transmission in Cooling Air with Extended Surfaces, by W. L. Knaus (*Refrigerating Engineering*, January, 1935, p. 23, February, 1935, p. 82).

[7]Graphical Method of Determining Finned Coil Capacities Described, by E. P. Wells (*Heating, Piping and Air Conditioning*, December, 1936, p. 665).

[8]Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall (*Refrigerating Engineering*, October, 1935, p. 211).

Table 3—Equations
(See List of Symbols, Table 8)

Eq. (A): Total heat transfer = overa l coefficient × total surface area × log mean temperature difference.

$Q_T = U \times (A \times \text{depth of coil} \times \text{face area}) \times mtd_1$

or,

$Q_T = h \times (A \times \text{depth of coil} \times \text{face area}) \times mtd_2$

where mtd_1 is measured from fluid to fluid, and mtd_2 is measured from fluid to surface.

Eq. (B): Sensible heat transfer = weight of air × specific heat × temperature change.

$Q_s = G \times \text{face area} \times 0.24 \times (T_1 - T_2)$

$Q_s = 0.24 \times \text{cfm} \times 60 \times 0.075 \times (T_1 - T_2)$

$Q_s = 0.24 \times \text{face velocity} \times \text{face area} \times 60 \times 0.075 \times (T_1 - T_2)$

Eq. (C): Total heat transfer = weight of air × change in enthalpy.

$Q_T = G \times \text{face area} \times (H_1 - H_2)$

$Q_T = \text{face velocity} \times \text{face area} \times 60 \times 0.075 \times (H_1 - H_2)$

Eq. (D): $Q_T = h_r/R\ (T_s - T_r)\ AN$

or,

$$T_s = T_r + \frac{RQ_T}{h_r AN}$$

Eq. (E): $1/U = 1/h_s + R/h_r$ ..... (Appendix 3)

Eq. (F): $h_s = (0.24G/AN)\ \log_e \dfrac{T_1 - T_s}{T_2 - T_s}$ ..... (Appendix 2)

others. (Table 1, method No. 5.) Usually a *mean* surface temperature is used, which is supposed to take care of the temperature gradient in the fins.

In this paper, surface temperature has been considered as the temperature of an imaginary surface having the same characteristics for heat transfer as the coil itself. To obtain a clearer idea of this interpretation, assume that the heat transfer coefficients have been determined for the coil under consideration. Then imagine the whole coil to be a thin-walled pipe or bundle of tubes having the same heat transfer coefficients as the coil, but consisting of prime surface only. Consider a refrigerant to be inside this pipe, and air to be blowing across the pipe on the outside. It is the average temperature of this type of imaginary surface that is called surface temperature, T_s in the development which follows.

Surface coefficients are usually determined from overall coefficients by some direct or indirect use of surface temperature in heat transfer equations, as in the method developed by Goodman.[9] However they are determined they may be expressed in terms of the velocity of the fluid with which they are associated, and they may be used to calculate the overall coefficients for any combination of conditions, by equation E, (Table 3). When calculations of the overall coefficient are made for a dehumidifying coil, due account must be taken of the latent heat as well as the sensible heat transfer.

The refrigerant-side surface coefficient is independent of what happens outside the coil; hence it is no doubt identical for both dry-cooling and dehumidification. The temperature gradient (or temperature rise) across the inside surface film is directly proportional to the total load; hence it changes when dehumidification begins, as does also the temperature gradient in the fins. The outside surface coefficient and the temperature gradient across the air film are both changed by the presence of dehumidification. Methods for taking these changes into account, and for determining the surface coefficients are given in the following section.

[9]Loc. Cit. See Note 4.

Derivation and Use of the Proposed Humidity Method

Before indicating the derivation of method No. 7, Table 1, it is well to review the characteristics of dehumidifying coils as indicated by the heat-load ratio lines on a psychrometric chart. If the inlet and exit air conditions are located on a non-logarithmic psychrometric chart (for example, points 1 and 4, Fig. 1) the angle which the straight line connecting the two points makes with the horizontal is a direct measure of the relative amounts of sensible heat and of latent heat extracted from the air by the coil, *i.e.*, this angle is a measure of the heat-load ratio. Conversely, if the initial condition of the air is located on the chart, and the required heat-load ratio line drawn through it, the exit condition will be somewhere on this line, near the saturation curve. In this latter case, the problem is to locate the exit point, usually by means of another line which will intersect the load-ratio line. The exit air point may thus be located as follows:

1. By finding the dry-bulb temperature of the exit air (Method No. 1, Table 1).
2. By finding the wet-bulb temperature of the exit air.
3. By finding the dew-point temperature of the exit air (Method No. 4).
4. By finding the relative humidity of the exit air.
5. By finding the wet-bulb depression (below dry-bulb) for the exit air.
6. By finding the ratio: Heat removed/heat removable = $\dfrac{\text{Length of line from inlet to exit air conditions}}{\text{Length of line from inlet to saturation curve}}$ (Method No. 5)

The method now to be described is approximately equivalent to finding the relative humidity of the exit air.

Suppose air enters a cooling coil at conditions corresponding to point 1 in Fig. 1. As long as the surface temperature of the coil is above the dew-point, the air will be cooled without loss of moisture, and its condition as it leaves the coil will be somewhere along the line 1-2. Its exact position on this line will depend on the air velocity and the air-side surface coefficient. When the surface temperature just equals the dew-point, the air will leave with conditions represented by point 2. But if the surface temperature is reduced below the dew-point, condensation will take place, and *the air will have a final condition somewhere along the line 2-3-4, which is a curve at a constant horizontal distance from the saturation curve. It must be understood that the path 1, 2, 3, 4, is not intended to represent the path of the condition of the air as it passes through the coil, but rather the path traced by the exit air conditions as the surface temperature is gradually reduced, with other conditions remaining constant.* For proof of the statement that the exit conditions lie along this line, see the section on Experimental Work, and also Appendix 1.

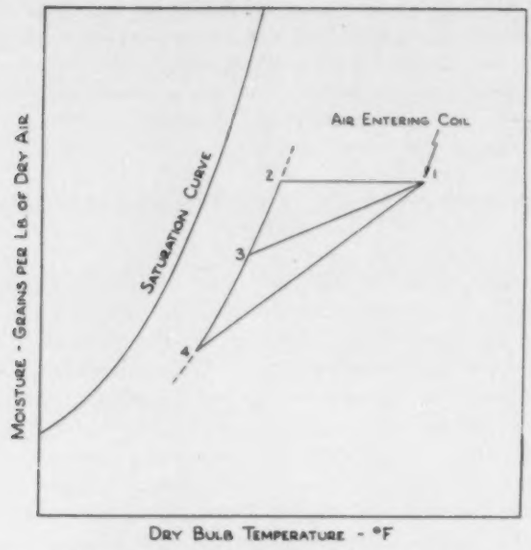

Fig. 1—Sketch of Humidity Method on psychrometric chart

The horizontal distance from the saturation curve to a given relative humidity line on the psychrometric chart is very nearly constant.[10] Therefore the minimum dry-bulb temperature for zero dehumidification (represented by point 2 in Fig. 1) and the exit conditions for any heat-load ratio, should have approximately the same relative humidity. Hence, method No. 7, Table 1, has been named the relative humidity method, or merely the *humidity method*.

Summary of Procedure for Humidity Method

1. Solve Eq. F, Table 3 for the minimum exit dry-bulb temperature which can be attained without dehumidification, and establish the intersection on the psychrometric chart of this temperature and the horizontal through the entering dew-point.
2. Through the point just established, draw a line of constant relative humidity. The intersection of this line with *any* load-ratio line determines the exit air conditions for that heat load ratio.
3. Solve Eq. B, Table 3, for face area.
4. Find surface temperature by Eq. F.

There are several other types of problems which are important in fin-coil rating or selection. Most of these are included in the summary of Table 4, which indicates the application of the *humidity method* to five other groupings of the knowns and unknowns besides the one used for Table 1. Trial-and-error solutions are not necessary in any of these except No. 1.

The proposed method has certain limitations with respect to the refrigerant conditions. Since a single surface temperature is used, the accuracy of the method is uncertain when the temperature rise of the water is large, as it may be in deep counterflow coils. With a direct-expansion refrigerant, the method of finding surface coefficients outlined in Appendix 4, is not suitable. One way to overcome this difficulty would be to test a similar coil with water refrigerant, and then use the same air-side surface coefficients for the direct-expansion coil.

A special advantage of the *humidity method* is that it

[10]See Relation of Dew-Point to Relative Humidity, Heating, Ventilating, Air Conditioning Guide 1938, Chapter 1, p. 9.

Table 4—Procedure for Solving Problems by Method No. 7[a]

	Problem		Procedure for Solution
	Given	Required	
1	Ent. Cond. Air velocity. Refrigerant velocity. Refrigerant temperature.	Exit Cond. Load ratio. Total heat transfer.	1. Find minimum dry-bulb for zero dehumidification by Eq. F with T_s = Entering DP. 2. Find T_s from Eq. D. 3. Find exit DB from equation F. 4. Draw a relative humidity line through the entering DP and DB found in (1). At the intersection with the DB found in (3), read exit WB. 5. Find total heat transfer from Eq. C. 6. Find sensible heat transfer and load ratio.
2	Ent. Cond. Air velocity. Refrigerant velocity. Load ratio.	Exit Cond. Refrigerant temperature. Total heat transfer.	1. Find minimum dry-bulb for zero dehumidification by Eq. F with T_s = Entering DP. 2. Draw a relative humidity line through the entering DP and DB found in (1). At the intersection with the load ratio line, read exit conditions. 3. Find total heat transfer from Eq. C. 4. Find T_s by Eq. F and T_r by Eq. D.
3	Ent. Cond. Air velocity. Load ratio. Refrigerant temperature.	Exit Cond. Refrigerant velocity. Total heat transfer.	1. Find minimum dry-bulb for zero dehumidification by Eq. F with T_s = Entering DP. 2. Draw a relative humidity line through the entering DP and DB found in (1). At the intersection with the load ratio line, read exit conditions. 3. Find total heat transfer from Eq. C. 4. Find T_s by Eq. F. 5. Find h_r/R by Eq. D and calculate V_r.
4	Ent. Cond. Refrigerant velocity. Total heat transfer. Exit Cond.	Load ratio. Refrigerant temperature. Air velocity.	1. Find air velocity by Eq. C. 2. Calculate sensible heat transfer and load ratio. 3. Find T_s by Eq. F. 4. Find T_r by Eq. D.
5	Ent. Cond. Exit Cond. Load ratio. Total load. Air velocity. Face area. Refrigerant velocity (or surface coefficient).	Coil depth. Refrigerant temperature.	1. Draw a relative humidity line through the exit conditions, and at the intersection of this line with a straight line through the entering DP, read the minimum DB for zero dehumidification. 2. Calculate h_a from air velocity and solve Eq. F for number of rows of coil depth. (N). 3. Solve Eq. F for T_s. 4. Find T_r from Eq. D.

[a]Problems 1 to 4 show how to find the capacity of a given coil. Problem 5 shows how to select a coil size for a given service.

gives a solution for cases in which the latent heat load is high. In those methods which use the intersection of the heat-load ratio line with the saturation curve as the surface temperature (see method No. 5, Table 1), it may be impossible to locate this intersection when the latent heat load is great.

Experimental Work

In order to examine the validity of the method just described, and of the derivations given in the Appendix, three fin-coils of radically different designs were tested.

The test equipment and the methods used were similar to those described in a previous report.[11] A line diagram of the test unit is shown in Fig. 2, and Fig. 3 is a photograph of the set-up.

The equations obtained for surface coefficients by analysis of the tests on the three designs of coils are given in Table 5. These were dry-coil tests. The method for finding the surface coefficients was suggested by McAdams,[12] and has been used by Pownall[13] and others. The procedure is outlined in Appendix 3. Air-side coefficients are expressed as a function of air velocity, and water-side coefficients as a function of water velocity.

In addition to the dry-coil tests for determining surface coefficients, many dehumidifying tests were made. For

[11]Loc. Cit. See Note 3.
[12]Heat Transmission, by W. H. McAdams, p. 264.
[13]Loc. Cit. See Note 8.

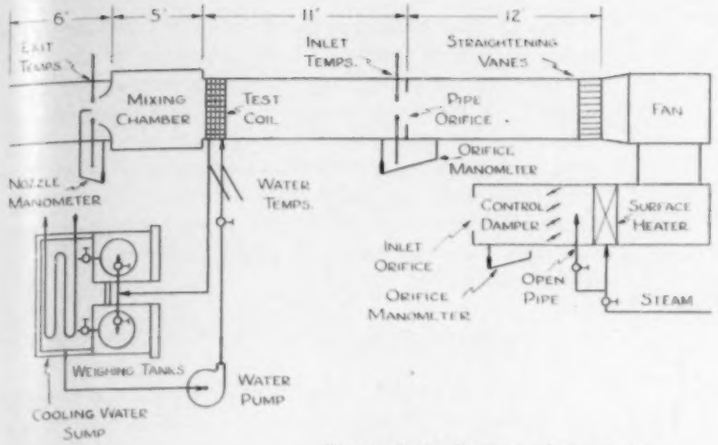

Fig. 2—Line diagram of test unit

each test condition a comparison was made between the calculated relative humidity of the exit air for maximum cooling without dehumidification, and the measured relative humidity at exit during the actual test. In other words, the relative humidities at points 2 and 3, Fig. 1, were compared. These comparisons are shown in the last two columns of Table 6. It will be noted that the agreement is very good in spite of a wide variety of operating conditions. While three different makes of coils are represented in Table 6, they were all 4-row cross-flow coils, and in order to examine the results with other coil depths, a number of the tests reported by Goodman[14] were analyzed in a similar manner (see Table 7). The validity of both Goodman's method and the *humidity method* is indicated by the fact that those test points which fell on his curves also showed a good check by the humidity method, but the points which departed from his curves failed to give an agreement between calculated humidity and test results.

Limitations of All Coil-Rating Methods

There is some question whether the actual performance of a dehumidifying coil under seemingly identical conditions is consistent within ± 5 per cent, and therefore whether *any* method can predict coil performance much closer than this. It has been noted before that a wet-coil test may be repeated under identical conditions of entering air temperatures and velocity, identical entering refrigerant temperature and velocity, and identical test set-up, and the coil capacity may vary 5 per cent, even when the heat-balance checks closer than 2 per cent.

It was suggested in a previous paper,[15] that these variations are due to differences between the latent-heat coefficients, depending on the predominance of drop-condensation or film-condensation. Schmidt and others[16] have reported differences greater than 400 per cent in

[14]Loc. Cit. See Note 4.
[15]Loc. Cit. See Note 3.
[16]Loc. Cit. See Note 12, p. 257.

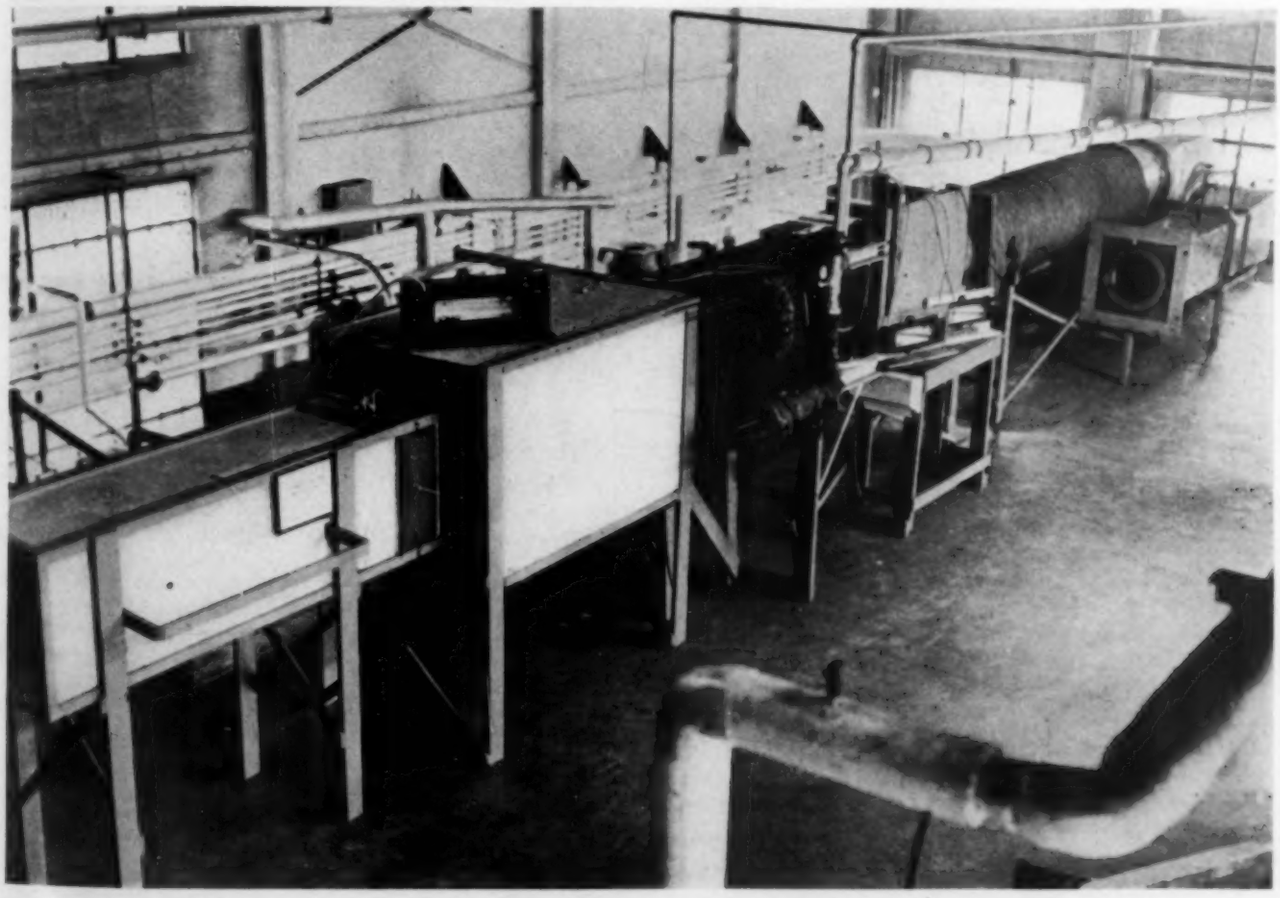

Fig. 3—Photograph of test set-up

Table 5—Surface Coefficients

Coil	Surface Coefficient	
	Air Side (h_a)	Refrigerant Side (h_r/R) Referred to Air Side Area
A	$0.0588\ V_a^{0.68}$	$17.55\ V_r^{0.8}$
B	$0.405\ V_a^{0.47}$	$22.7\ V_r^{0.8}$
C	$0.130\ V_a^{0.76}$	$21.4\ V_r^{0.8}$

Table 6—Comparison of Test Results with Computed Results

Coil	Test No.	Load Ratio $\frac{S}{T}$	Total Load Btu/Hour Sq Ft Face	Entering Conditions		Air Vel. Fpm	Relative Humidity	
				Dry-Bulb	Wet-Bulb		At Exit (By Test)	At Td (Computed)
A	15	0.638	11,320	83.9	74.0	477	90.0	90.5
	16	0.444	13,000	86.0	78.5	324	92.0	93.0
	17	0.534	13,830	86.2	76.6	415	91.0	91.0
B	5	0.516	9,350	83.0	75.0	273	93.0	94.0
	6	0.302	9,430	75.3	75.3	271	99.5	100.0
	12	0.634	10,050	84.1	74.4	540	92.0	93.0
	13	0.895	15,860	102.0	82.5	685	83.0	83.0
	32	0.638	13,920	91.7	79.5	578	90.5	90.5
	83	0.802	10,090	83.9	72.0	574	89.0	89.0
	34	0.795	11,600	85.4	72.6	571	89.0	89.0
	36	0.703	9,870	84.5	74.1	463	91.0	91.0
	37	0.809	8,620	85.4	74.0	474	90.0	90.0
	38	0.894	9,610	81.3	70.1	635	88.0	88.0
	39	0.596	14,100	83.5	75.5	635	92.0	92.0
	40	0.652	9,280	77.1	70.5	632	93.0	93.0
	41	0.608	11,200	91.8	77.8	274	93.0	93.0
C	29	0.670	12,920	88.8	73.1	389	79.5	80.0
	10	0.730	17,160	100.0	78.0	363	80.5	79.5
	15	0.440	16,300	85.7	77.9	391	89.5	90.5
	11	0.650	18,000	100.0	80.1	369	82.0	82.0
	30	0.640	17,100	89.8	76.0	483	84.0	85.0
	8	0.460	19,500	89.9	79.8	399	87.0	89.0
	25	0.735	12,400	90.0	72.4	295	83.0	83.0
	6	0.350	14,000	79.8	76.1	385	95.0	96.0

Table 7—Experimental Results by Goodman Recalculated by Humidity Method.[a] (See Reference No. 4.)

Test No.	No. of Rows	Load Ratio	Entering Conditions			Relative Humidity	
			Dry-Bulb	Wet-Bulb	Air Velocity Fpm	By Test Per Cent	Calculated Per Cent
The following points fall close to Goodman's condition curves:							
16	4	0.795	84.0	68.4	654	80.5	80.5
16	5	0.775	84.0	68.4	651	87	87
16	6	0.760	84.0	68.4	654	89	89.5
16	7	0.755	84.0	68.4	654	93	93
17	5	0.500	85.9	76.1	694	92	92
17	6	0.480	85.9	76.1	694	91	93
18	5	0.600	87.4	74.2	689	88	88
18	6	0.580	87.4	74.2	689	89	91
25	5	0.900	106.1	75.1	702	75	75
These points *do not* fall close to Goodman's condition curves:							
17	7	0.462	85.9	76.1	694	89	95
18	7	0.560	87.4	74.2	689	90	94
25	7	0.825	106.1	75.1	702	80	85

[a]These data were based on Table 7 in Goodman's paper.

coefficients of heat transfer for steam condensation on surface-condenser tubes with identical operating conditions but different previous history of the tube-surface.

Another difficulty in all coil tests is the near impossibility of securing sufficiently thorough mixing of the exit air so that temperatures can be measured to one or two tenths of a degree. The difficulty is especially great with wet-bulb readings. The type of mixing-box specified in the Standard Code for Testing and Rating Steam Unit Heaters of the American Society of Heating and Ventilating Engineers and the Standard Method of Rating and Testing Air Conditioning Equipment of the *American Society of Refrigerating Engineers* are very unsatisfactory in this regard, and suggestions for improvement are solicited.

Still another limitation on the accuracy of methods of rating dehumidifying coils is the presence of the water of dehumidification. Many designs of coils do not drain steadily, or the exact nature of the condensate film depends on just how the coil is installed, upon its cleanliness, and upon its immediate past history. All of these limitations call for additional attention to the problem of the inherent consistency and reproducibility of wet-coil operation, and hence the probable limits of accuracy of coil ratings.

Suggested Program for Fin-Tube Research

In order to develop an acceptable uniform method for rating and comparing dehumidifying coils and for calculating their performance under a variety of conditions, the following program for the continuance of this research should be carried out as rapidly as possible:

1. *Variation of air-side surface coefficients.* The variation of the air-side coefficient should be studied, especially with reference to the comparison of dry- and wet-coils, and at various heat-load ratios. These determinations will furnish a further check on the accuracy of the *humidity method* for obtaining coil performance.

2. *Refrigerant-side coefficients for water.* The effect of water velocities on the water-side coefficients should be further studied, especially for the higher water velocities now becoming common. The actual advantages of the counterflow arrangement as compared with cross-flow should be demonstrated by comparative tests.

3. *Refrigerant-side coefficients for direct expansion.* For dichlorodifluoromethane in particular, a study of refrigerant-side coefficients for approved methods of coil arrangement will clear up some of the uncertainties which exist in the comparisons between direct-expansion and water coils. It will also give definite information regarding the amount of dehumidification possible at various low-side pressures (or refrigerant temperatures). Suitable experimental methods for dichlorodifluoromethane-coil tests should be developed and specified.

4. *Depth of coils.* The effect of the depth of coil upon its heat transfer characteristics should be studied further, such as by tests on coils 2, 4, 6, 8, and 10 rows deep.

5. *Relation of heat transfer to air velocity and friction.* Further studies should be made of the economics of coil design and selection with respect to heat transfer and air friction.

The suggested program should cover both continuous-fin and round-fin coils.

All work should be directed toward the setting up of uniform Test Code methods for rating and calculating surface coils. The accuracy of the various methods should be studied further.

Appendix No. 1

Proving that: *The locus of the exit air conditions from a coil, as refrigerant temperature is varied, is a line on the psychrometric chart at a constant horizontal distance from the saturation curve.*

The proof of the method proposed for analysis of wet-coil performance will now be given. For this purpose it will be necessary to derive an expression for heat load ratio in terms of entering air conditions and refrigerant temperature.

From the psychrometric chart, the moisture content corresponding to any dew-point may be found in terms of temperature if the equation of the saturation curve is known in terms of moisture content and temperature. However, a simpler method for finding moisture content in terms of temperature can be used in this derivation.

Referring to Fig. 4, if a straight line AB is drawn through

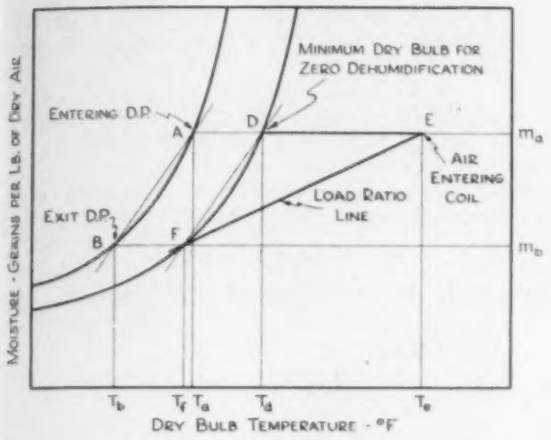

Fig. 4—Development of Humidity Method

the points on the saturation curve which represent the entering and final dew-points of air being cooled by a dehumidifying coil, the values of the moisture content at these dew-points may be expressed in terms of the slope and intercept of that line. The general expression is: (See List of Symbols, Table 8.)

$$m = dm/dT\ (T) + C \quad \ldots\ldots (1)$$

Therefore, the moisture difference between the entering and final conditions may be represented by the following expression which was obtained directly from equation (1) by solving for m_a and m_b and then subtracting the results.

$$m_a - m_b = dm/dT\ (T_a - T_b) \quad \ldots\ldots (2)$$

where the subscripts a and b refer to any dew-points whatever. The slope dm/dT depends on the actual values of T_a and T_b, that is, the slope of the straight line connecting the initial and final dew-points varies with the position of those dew-points on the saturation curve.

The line EF in Fig. 4 is the load-ratio line corresponding to any coil performance in which the air enters the coil with the moisture content m_a, and leaves the coil with the moisture conten m_b. The latent heat extraction through the coil is:

$$Q_L = \frac{L}{7000}(m_a - m_b)$$

$$= \frac{L}{7000}(m_a - m_s)\ f\ (V_a) \quad \ldots\ldots (3)$$

where $(m_a - m_s)$ equals all the moisture which would be abstracted from the air if it were cooled down to the coil surface temperature, and $f(V_a)$ takes care of the fact that it is cooled only a portion of this distance. In other words, the ratio of the latent heat removed to the maximum removable with the given surface temperature is a function of the air velocity (and, of course, of the coil design and coil depth).

Replacing $(m_a - m_b)$ in Equation (3) by its equivalent from Equation (2), the latent heat transfer may be expressed in terms of temperatures rather than moisture contents:

$$Q_L = \frac{L}{7000}\frac{dm}{dt}(DP - T_s)\ f\ (V_a) \quad \ldots\ldots (4)$$

It has been assumed by several investigators[17] that the ratio of heat removed to heat removable is similarly affected by air velocity, whether latent heat, sensible heat or total heat is being considered. Making this assumption, the following may be written:

$$Q_s = 0.24\ (T_1 - T_s)\ f(V_a)$$

But:

$$Q_L + Q_s = Q_T;\ \text{or}\ \ Q_T/Q_s = Q_L/Q_s + 1$$

Therefore:

$$Q_T/Q_s = \frac{L\ dm/dt\ (DP - T_s)\ f\ (V_a)}{0.24 \times 7000\ (T_1 - T_s)\ f\ (V_a)} + 1 =$$

$$\frac{K\ dm/dt\ (DP - T_s)}{(T_1 - T_s)} + 1 \quad \ldots\ldots (5)$$

From which:

$$T_s = \frac{K\ dm/dt\ DP\ -\ T_1\ (Q_L/Q_s)}{K\ dm/dt - Q_L/Q_s} \quad \ldots\ldots (6)$$

Now, select any set of entering conditions and hold them constant (as well as the air velocity and refrigerant velocity) and let the surface temperature be varied by variation of refrigerant temperature. Under these conditions, decreasing surface temperatures will result in increasing latent heat transfer, and the result on a psychrometric chart will be similar to Fig. 1: where line 2, 3, 4 is a curve at a constant horizontal distance from the saturation curve. This statement will now be proved by substituting different values of surface temperature in Equation F. (See appendix No. 2.)

$$\frac{h_a AN}{0.24\ G} = log_e \frac{T_1 - T_{s1}}{T_2 - T_{s1}} = log_e \frac{T_1 - T_{s2}}{T_2 - T_{s2}}$$

The assumption that h_a for sensible cooling is the same for both wet and dry surface has been made by other investigators[18] and is required here.

Table 8—List of Symbols

$dm/dT =$	Slope of straight line connecting initial and final dew-points, (on the saturation curve).
$m =$	Moisture content in grains per pound of dry air.
$m_a = m_2 =$	Moisture content at A in grains per pound of dry air.
$m_b = m_3 =$	Moisture content at B in grains per pound of dry air.
$T =$	Dry-bulb temperature.
$T_a =$	Dry-bulb temperature (and dew-point) at A.
$T_b =$	Dry-bulb temperature (and dew-point) at B.
$T_1 = T_e =$	Entering dry-bulb temperature.
$T_2 = T_d =$	Minimum dry-bulb temperature for zero dehumidification.
$T_3 = T_f =$	Exit dry-bulb temperature.
$T_r =$	Refrigerant temperature.
$T_s =$	Surface temperature.
$Q_s =$	Sensible heat transfer.
$Q_L =$	Latent heat transfer.
$Q_T =$	Total heat transfer.
$DP =$	Entering dew-point.
$L =$	Latent heat of vaporization.
$K =$	L/0.24 × 7000.
$G =$	Weight of air in pounds per hour per square foot of face area.
$h_a =$	Air side surface coefficient in Btu per hour per square foot surface per degree, sensible heat.
$V_a =$	Face velocity, feet per minute.
$h_r =$	Refrigerant side surface coefficient in Btu per hour per square foot surface per degree.
$V_r =$	Refrigerant velocity, feet per second.
$R =$	Ratio of air side to refrigerant side surface.
$A =$	Air side surface area in square feet per square foot of face per row coil depth.
$U =$	Overall coefficient in Btu per hour per square foot surface per degree.
$N =$	Number of rows of coil depth.

Then,

$$\frac{T_1 - T_{s1}}{T_2 - T_{s1}} = \frac{T_1 - T_{s2}}{T_2 - T_{s2}} \quad \ldots\ldots (7)$$

[17] Loc. Cit. See Notes 6 and 7.

[18] Loc. Cit. See Note 4.

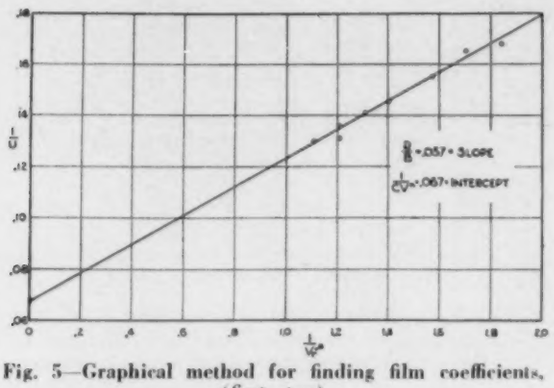

Fig. 5—Graphical method for finding film coefficients, (first step)

Substituting (6) in (7) and letting $X = Kdm/dT$ and $Z = Q_L/Q_S$

$$\frac{T_1 - \dfrac{X\,DP - Z_1T_1}{X - Z_1}}{T_2 - \dfrac{X\,DP - Z_1T_1}{X - Z_1}} = \frac{T_1 - \dfrac{X\,DP - Z_2T_1}{X - Z_2}}{T_2 - \dfrac{X\,DP - Z_2T_1}{X - Z_2}}$$

$$\frac{XT_1 - Z_1T_1 - X\,DP + Z_1T_1}{XT_2 - Z_1T_2 - X\,DP + Z_1T_1} = \frac{XT_1 - Z_2T_1 - X\,DP + Z_2T_1}{XT_2 - Z_2T_2 - X\,DP + Z_2T_1}$$

Collecting terms and dividing both sides by $(XT_1 - X\,DP)$,

$XT_2 - Z_1T_2 + Z_1T_1 - X\,DP = XT_2 - Z_2T_2 + Z_2T_1 - X\,DP$

or, $XT_2 - Z_1T_2 + Z_1T_1 = XT_2 - Z_2T_2 + Z_2T_1$ ..... (8)

But, $0.24\,G(T_1 - T_2) = Q_{S1}$

Therefore, $T_2 = T_1 - Q_{S1}/0.24\,G$; and $T_2 = T_1 - Q_{S2}/0.24\,G$.. (9)

Substituting (9) in (8),

$XT_1 - XQ_{S1}/0.24\,G - Z_1T_1 + Z_1Q_{S1}/0.24\,G + Z_1T_1 =$
$XT_1 - XQ_{S2}/0.24\,G - Z_2T_1 + Z_2Q_{S2}/0.24\,G + Z_2T_1$

Collecting terms and multiplying by $0.24\,G$,

$Z_1Q_{S1} - XQ_{S1} = Z_2Q_{S2} - XQ_{S2}$ ..... (10)

But, $Z_1 = Q_{L1}/Q_{S1}$; $Z_2 = Q_{L2}/Q_{S2}$ and $X = Kdm/dT$ and $K = L/0.24 \times 7000$

Substituting these values in (10)

$Q_{L1} - (L/0.24 \times 7000)\ (dm/dT)\ Q_{S1} = Q_{L2} - (L/0.24 \times 7000)\ (dm/dT)\ Q_{S2}$

Therefore,

$(L/0.24 \times 7000)\ (dm/dT) = (Q_{L1} - Q_{L2}) / (Q_{S1} - Q_{S2}) =$
$L\ (m_1 - m_2 - m_1 + m_2)/0.24 \times 7000\ (T_1 - T_2 - T_1 + T_2)$

or, $dm/dT = (m_2 - m_5)/(T_2 - T_2)$ ..... (11)

Now, points 2 and 3 correspond to points D and F in Fig. 4. Therefore, line DF has a slope equal to dm/dT which has been assumed constant and equal to the slope of AB. This assumption is a close approximation for that part of the saturation curve used in comfort air conditioning.

Therefore, it has just been proved that the line connecting point D to point F in Fig. 4 has the same slope as the line AB, connecting the initial and final dew-points on the saturation curve. But D and F have the same dew-points as the initial and final conditions, (A and B). Therefore, a line with the same slope as AB drawn through D must intersect a curve through D and at a constant horizontal distance from the saturation curve, precisely at F. But in the operating range of comfort air conditioning, a line of constant relative humidity almost exactly meets this description. Hence, the curve 2, 3, 4 in Fig. 1 (or the curve DF in Fig. 4) may be taken as a line of constant relative humidity. Since the load ratio line is a line which connects initial and final conditions, it also must pass through F. Therefore, point F can be determined from the intersection of the line of constant relative humidity through D, and it is not necessary to find the slope of the straight line connecting D and F.

Since F was selected as a perfectly arbitrary point, its position can always be found by intersection of a relative humidity line and the load ratio line as just explained. Therefore, a relative humidity line through point D must be the locus of all exit air conditions as the surface temperature is gradually decreased, other conditions remaining constant.

Appendix No. 2

Derivation of exit dry-bulb equation. (Equation F, Table 1.)

$$Q_S = h_aA(MTD)N$$

$$Q_S = \frac{h_aA(T_1 - T_2)N}{\log_e \dfrac{T_1 - T_S}{T_2 - T_S}}$$

But, $Q_S = 0.24\,G(T_1 - T_2)$

Therefore, $0.24\,G(T_1 - T_2) = \dfrac{h_aA(T_1 - T_2)N}{\log_e \dfrac{T_1 - T_S}{T_2 - T_S}}$

Or, $Nh_aA/0.24\ G = \log_e\ (T_1 - T_S)/(T_2 - T_S)$.

Appendix No. 3

Film Coefficients

The suggested method for finding film coefficients involves the use of two curves shown by Figs. 5 and 6. Both of these curves may be obtained from dry-coil tests only.

Fig. 5 is an application of a method of finding the inside film coefficient as suggested by McAdams. The curve may be obtained from a series of dry tests run at constant air velocity and variable refrigerant velocity. Since $1/U = 1/h_a + 1/h_r$, the following expression may be written by replacing the h values in terms of velocity: $1/U = 1/CV_a^n + R/BV_r^{0.8}$; where C and B are constants to be determined, R is the ratio of external to internal surface, n is an exponent to be determined, and V_a and V_r are the air and refrigerant velocities respectively. Now if tests are run at constant air velocity and variable refrigerant velocity, values of $1/CV_a^n$ will remain constant, and the equation reduces to the general equation of a straight line in which the value of $1/CV_a^n$ will be the zero intercept and the value of R/B will be the slope. (See Fig. 5.)

Now, if tests are run at constant refrigerant velocity and variable air velocity, $R/BV_r^{0.8}$ will be a *known* constant, and the equation will reduce to the following form: $1/U - R/BV_r^{0.8} = 1/CV_a^n$. This is the equation of a straight line on log paper, and the intercept will give the value of $1/C$ while the slope will give the exponent n. (See Fig. 6.)

After Figs. 5 and 6 have been drawn, all the constants in the equation $1/U = 1/CV_a^n + R/BV_r^{0.8}$, will be known, and either the film coefficients or the value of the overall coefficient *for dry cooling* may be calculated for any set of conditions. These dry cooling coefficients may then be applied to cooling in which dehumidification takes place.

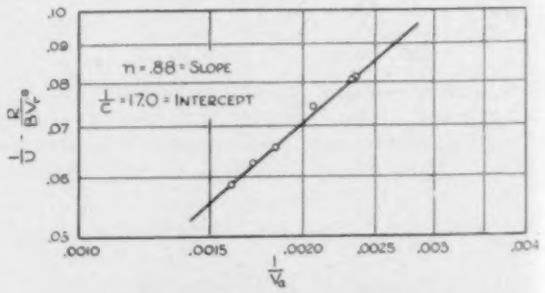

Fig. 6—Graphical method for finding film coefficients, (second step)

A.S.H.V.E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers*

COMMITTEE: C. E. Bronson, *Chairman*; L. A. Harding, H. M. Hart, F. C. Houghten, A. J. Johnson, J. F. McIntire, D. W. Nelson, Percy Nicholls, R. A. Sherman, and E. C. Webb.

I. INTRODUCTION

1. **The purpose** of the Code for Testing Stoker-Fired Steam-Heating Boilers is to provide a standard method for conducting and reporting tests to determine heat efficiency and performance characteristics; this Code does not fix the conditions that must be fulfilled for the acceptance or approval of the combination of boiler and stoker. The Code recognizes that tests of stokers and boilers may be made under different conditions for different purposes, and with complete or limited facilities for conducting the tests.

It should be appreciated that in all cases to which this Code applies there are two devices, that is, boiler and stoker, involved in the test. Both devices influence the results obtained. Two different stokers placed successively in the same boiler may not produce identical results. To view the results as if they pertained only to the boiler would be an incorrect viewpoint.

2. **Standard Form.** Table 1, Standard Form, Solid Fuels —Stoker Fired is arranged for recording the full data for a continuous test called for by this Code.

3. **Short-Test Form.** Table 2, Short-Test Form, Solid Fuels, Stoker Fired is for intermittent tests or for continuous tests in which a complete heat balance is not desired. The following determinations are not required:

(a) Proximate analysis of fuel, except as required for record.
(b) Ultimate analysis of fuel.
(c) Complete flue gas analysis; CO_2 is required for record.

4. **Continuous and Intermittent Tests.** Two general types of tests are recognized; they will be designated as Continuous and Intermittent. A Continuous test may be defined as one where the stoker operates at a substantially constant rate throughout the duration of the test; it may use either the Standard or Short-Test Form. An Intermittent test is one where the stoker either shuts off or operates at a lower rate at intervals throughout the test; it is designed to cover the determination of only the stoker and boiler efficiency and may use only the Short-Test Form.

II. DEFINITIONS

5. **Furnace Volume** is the space which may be used for combustion. It shall be considered as that space enclosed by the hearth or grate surface and the side walls or water legs up to the crown sheet or equivalent. If a bridge wall is used, that space back of the bridge wall above the horizontal plane passed through the top of the bridge wall shall be included.

6. **Hearth or Grate Area** is the projected area bounded by the furnace walls at the hearth or grate level except for spillover stokers where it is the projected area of the retort.

7. **Combustion Rate** is the number of pounds of fuel, as fired, fed by the stoker per hour, less the fuel equivalent of the combustible in the ash. The fuel equivalent of the combustible in the ash shall be taken as the weight of the combustible multiplied by the ratio of 14,600 to the calorific value of the fuel as fired.

8. **Heat Release Rate,** in Btu per hour, is the product of the combustion rate and the calorific value of the fuel.

9. **Boiler Output** in Btu per hour, is the quantity of heat delivered at the boiler nozzle with the boiler normally insulated.

10. **Efficiency of Stoker and Boiler** is the percentage of the heat in the fuel fired which is delivered at the boiler nozzle when the boiler is normally insulated.

III. SET-UP OF STOKER AND BOILER TO BE TESTED

11. **Stoker and Boiler** when set up for the purpose of commercial testing shall be assembled according to the manufacturer's directions.

12. **Setting** of the stoker in relation to the boiler shall be as agreed to by all interested parties. NOTE: The report of a test shall clearly describe the manner of installation.

13. **Boilers** already installed shall be arranged to comply with the following conditions as nearly as possible, or as shall be agreed to by all interested parties.

14. **Boiler Covering.** The boiler should be covered with such heat insulation as is recommended or commercially furnished by the manufacturer; if tested uninsulated or partly insulated, an allowance shall be made for that portion of the heat lost from the bare metal steam surface which might have been saved by covering. The allowance shall be computed from the formula given with the tables of this Code. In no case shall allowance be made for uninsulated portions which would also be uninsulated in the commercial product.

15. **Steam Outlet Piping.** All steam outlet pipes from the boiler shall always be well covered, up to and including the steam separator.

16. **The Vertical Outlets** from the boiler leading to the separator shall be in number, arrangement, and size according to manufacturer's directions and shall not rise vertically from the boiler nozzle more than 30 in. before entering the steam separator.

17. **The Steam Piping** leading from the separator shall have a slope so that water condensed in it will drain away from the separator. The drain pipe from the steam separator shall include a U-leg sufficiently long to prevent the steam blowing out, and with a short inverted U-leg at the outlet end so that the drip may be caught in a container. Re-evaporation of water drained from the separator shall be prevented by use of a suitable cover on the container.

18. **The Steam Valve** on the boiler steam outlet piping shall be placed beyond the steam separator.

19. **Feed Water Piping.** The feed water pipe should connect to the boiler where the returns would normally come. The temperature of the feed water shall be read from a ther-

*Adopted 1938 by the American Society of Heating and Ventilating Engineers, New York, N. Y., and the *Stoker Manufacturers Association*, Chicago, Ill.

mometer inserted into the feed line near the boiler but not so close that the temperature will be affected by heat conducted from the boiler. The thermometer may be in a well or cup, or it may be inserted directly into the feed water. The latter method is recommended to avoid errors introduced by heat conduction of a well.

A valve shall be placed on each side of the thermometer position. The valve between the thermometer and the boiler shall serve as a shut-off valve. The valve between the thermometer and feed water supply shall be used as a regulating valve only. It is recommended that this valve be an automatic one for maintaining a uniform water level in the boiler. If only one valve is used, it shall be placed between the thermometer and the boiler.

All boiler-water connections, including blow-off pipes, must be exposed to view, so that leakage may be observed and either stopped or measured.

20. Water Gage Marking. The water gage shall have a permanent mark to indicate the water level recommended by the manufacturer; if there is no permanent mark, for the purposes of these tests, a mark corresponding to the level recommended by the manufacturer shall be put on the water column gage glass.

21. Smokehood Connections. The boiler shall be connected with a short, direct smoke pipe to a source of draft. All joints shall be thoroughly sealed and maintained so during the tests.

The smokehood and the smoke flue shall be insulated with not less than 1 in. thickness of insulation; the smoke-flue insulation shall extend at least one equivalent flue diameter beyond the point where the flue-gas temperature is measured.

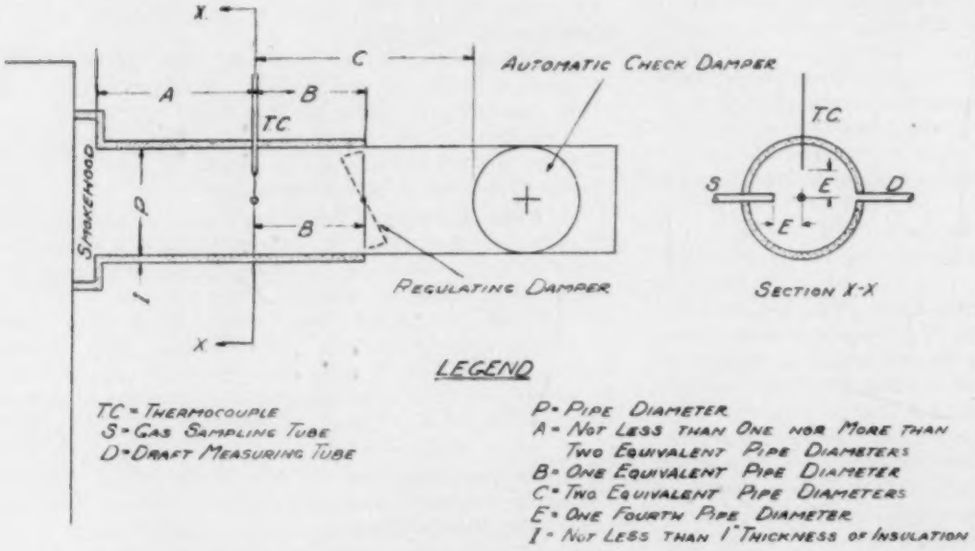

Fig. 1—Location of thermocouple, gas sampling tube, draft tube, and dampers

There shall be no check draft damper between the boiler and the point where the flue-gas sample is taken or the flue-gas temperature measured; if one is incorporated in the boiler, it shall be thoroughly sealed during all tests.

The choke damper, if incorporated in the boiler, shall be put in place, opened wide, and kept so during all tests. A damper shall be placed in the smoke pipe between the source of draft and the point where flue-gas measurements are made. This shall be called the regulating damper. A check damper, either manual or automatic, may be placed between this damper and the source of draft, if desired. An automatic damper is recommended. This check damper shall not be placed closer than two pipe diameters to the point where readings are being taken.

22. Source of Draft. The draft may be produced by a chimney or by a fan or other arrangement for providing induced draft.

23. Cleaning of Boiler. The water spaces of the boiler shall be thoroughly boiled out with a cleaning solution, then thoroughly rinsed with clean water.

24. Soot, Ashes, and Dust. The heating surface, furnace, and flues shall be clean and free from soot, ashes and dust at the beginning of the test.

IV. INSTRUMENTS AND MEASURING APPARATUS

25. Steam Output. The weight of water evaporated may be determined either by condensing the steam and weighing the condensate, designated as Method *A*, or by weighing the water fed to the boiler, designated as Method *B*.

Method A. Water fed to the boiler need not be weighed. The steam after passing through the separator shall be led to a condenser through a connecting steam pipe sloping toward the condenser. The condenser shall be so set up that all the condensate will drain quickly to the outlet. The condensate shall be caught in suitable tanks and weighed.

The condenser shall be of the closed type, preferably with outside joints, and of sufficient capacity that the temperature of the condensate shall not exceed 150 F. Before and after each test it shall be tested for freedom from leakage by putting it under full water pressure.

Method B. The weight of water fed to the boiler shall be determined in an accurate and suitable manner. The most convenient weighing arrangement depends on the size of the boiler, but it shall be such that the weight fed can be determined at any instant. The use of tanks fitted with calibrated floats or gage glasses is permissible.

The water may be fed to the boiler by gravity, air pressure, or feed pumps; any leakage between the measuring tanks and boiler shall be weighed and deducted from the water fed.

26. Weighing Scales. Accurate scales of suitable size shall be provided for weighing separator water, feed water, fuel, and all refuse removed from the furnace.

27. Draft and Pressures. A suitable number of draft gages shall be provided and so arranged as to determine the pressure difference between the room atmosphere and the stoker's windbox, the furnace, and the smokehood. Draft measurement shall be made with draft gages reading to 0.01 in. of water. Draft gages shall be checked for zero reading each hour.

28. The Smokehood Draft shall be measured at least one pipe diameter before the regulating damper. The connection into the flue shall have its end square and shall be set at right angles to the flow of the gases.

29. Furnace Draft. The connection for measurement of the furnace draft shall be made through a hole in the side or front of the furnace at a level approximately midway between top and bottom of the furnace door.

30. **Stoker Windbox Pressure.** The stoker windbox pressure shall be measured at a suitable point in the plenum chamber from which air is delivered to the grates or the tuyeres. The connection shall have its end square and flush with the inner wall of the plenum chamber and shall be so placed as to read the true static pressure.

31. **Steam Separator.** The steam separator shall have sufficient volume to catch slugs of water and to prevent their passing into the steam line; it shall also efficiently separate entrained water from the steam carrying it.

32. **Temperature Measurements.** Accurately calibrated instruments shall be provided for all temperature measurements. A mercury thermometer graduated to not more than 1 F is preferable for measuring feed water temperature. It is recommended that thermocouples, in preference to other types of thermometers, be used to measure the flue-gas temperatures. The temperature shall be measured in the smoke pipe not less than one nor more than two pipe diameters beyond the smoke-pipe collar. The thermo-junction or bulb of the thermometer shall be placed as shown in Fig. 1. The instrument used with the couple should be sensitive to at least 5 F and have a range up to 1200 F.

33. **Sampling and Analysis of Flue Gas.** The sample of flue gas for analysis shall be taken from the smoke pipe at the point at which the temperature is measured in such a manner as to give a fair average sample of the gas stream. An Orsat or equivalent gas analyzer shall be used.

An open end ⅛ in. iron pipe reaching one-fourth way across the flue pipe is usually satisfactory, but the probable mixing of the gases should be considered and the open end so placed as to be in the average gas stream. The use of a perforated pipe may be advisable in certain cases and shall be permissible. For gas temperatures of over 750 F the portion of the collecting pipe extending into the flue should be clay or silica, because the iron at this temperature may reduce CO_2 to CO. Air leakage around draft tubes, thermocouples, and gas-sampling tubes shall be prevented by using suitable packing or seal around the tubes.

As an average value for the test period is desired, the correct and simplest method is to collect samples of the flue gas at a constant rate into a bottle and to take the samples for analysis from those bottles.

If recording gas analyzing instruments are provided, they shall be checked every hour with the Orsat or other manual apparatus.

34. **Smoke Readings.** Smoke readings shall be made by the Ringelmann Chart Method.

35. **Pressure Readings.** A calibrated steam gage, a mercury column, or a mercury thermometer graduated to 1 F, and inserted directly into the steam space of the boiler shall be used for determining the steam pressure.

V. FUEL SAMPLING

36. **(a) As the hopper of the stoker** is filled during the test, samples shall be taken, in small increments, at regular intervals and stored in a covered water-tight container in a cool place. The sample shall be approximately 10 per cent of the fuel fired; the correct amount can readily be obtained by taking as a sample every tenth shovelful as the hopper is filled. In no event shall the total sample be less than 100 lb.

The gross sample shall be quickly reduced to a laboratory sample of 15 lb by the methods described in the current Standard Method of Sampling Coal of the *American Society for Testing Materials*, Designation D21, and placed in a sealed container for transportation to the laboratory.

(b) When a number of tests are to be conducted with the same lot of fuel, it shall be permissible to sample the entire lot for ultimate and proximate analysis and calorific value as specified in the current Standard Method of Sampling Coal of the *American Society for Testing Materials*, Designation D21. When this method is used, samples shall be collected during each test, as in Paragraph 36*a* for determination of moisture and ash, and the ultimate and proximate analyses and calorific value for the gross lot corrected for same, as applied to the individual tests.

37. **(a) After the ash and refuse** has been weighed, it shall be reduced to a 15 lb laboratory sample for determination of its combustible content.

(b) In stokers equipped with automatic ash removal mechanism, this shall be disconnected and the ash collected directly from the ashpit when the test is less than 24 hours in duration.

In stokers operating on the principle of removal of ash as clinker, only the fused clinker shall be removed for weighing, sampling, and analysis.

VI. DURATION OF TESTS

38. **The duration of tests** shall be based on the firing of the minimum quantity of fuel to make the quantitative errors comparable for all tests. For both continuous and intermittent tests this quantity shall be considered as 10 times the hourly *continuous* rate of feed of the stoker at the setting of the feed being used, but in no case shall the total quantity of fuel fired be less than 200 lb per square foot of hearth or grate area.

VII. STARTING AND STOPPING OF CONTINUOUS TESTS

39. **Before starting any test,** if a different type or lot of fuel from that used in the preceding test is to be used, the stoker hopper shall be emptied and the feeding mechanism shall be flushed free of previous fuel. The feeding mechanism may be considered cleaned as the result of having it convey a volume of new fuel equal to twice the estimated gross volume of the feeding mechanism.

40. **The fire shall be started** by filling the feeding system with fuel fed from the hopper. A suitable amount of kindling shall be placed on the fuel in the furnace and ignited. The air supply shall be adjusted as recommended by the manufacturer or as indicated desirable by experience.

41. **Operating Conditions.** After the stoker has been started and the prescribed test conditions of rate of feed, draft in furnace, and air supply have been adjusted, all operating conditions shall have remained reasonably constant for one hour before the test period is started. The preliminary period shall be not less than 4 hours' duration.

42. **(a) Start of Test.** In stokers with automatic ash removal, the ashpit or ash removal mechanism shall be thoroughly cleaned to remove ash and refuse from previous tests and from the preliminary period. In stokers operating on the principle of removal of ash as clinker, any clinker that shall have been formed shall be removed. At the end of the preliminary period and after cleaning the fire the thickness and condition of the fuel bed shall be observed and recorded.

(b) The time, feed water, and boiler gage glass level shall be noted and recorded. The fuel in the stoker hopper shall be leveled off and the height recorded. The test shall start at the time of making these observations. A pail shall be placed under the separator outlet.

43. **(a) End of Test.** At the end of the test the fuel bed shall be as closely as possible of the same thickness and condition as at the start. After the clinker has been removed, if the fuel bed should be heavier than at the start, it is permissible to reduce the coal feed or to increase the air supply for a time necessary to burn the bed down to the thickness at the start. If it is thinner, it is permissible to reduce the rate of air supply for the time necessary to build the fuel bed up.

(b) When the proper condition of the fuel bed is reached, the water level in the boiler and the level of the fuel in the stoker hopper shall be adjusted to the same height as at the

start; the time shall be recorded and this time shall be the time of stopping. The feed water shall be quickly measured and recorded, and the separator water in the pail shall be weighed and recorded.

(c) In stokers operating with continuous or automatic ash removal, the ash shall be removed and weighed before preparation of the sample.

VIII. STARTING AND STOPPING OF INTERMITTENT TESTS

44. (a) Measurement and Procedure. In general, the same measurements and procedure must be followed in these tests as outlined in Section VII. Due to the fact that intermittent operation includes a heating and cooling of the boiler, it is absolutely necessary to be sure the same conditions exist at both the beginning and end of the test.

(b) These tests shall be started and stopped at the end of an *on* period.

(c) The intervals of stoker operation may be controlled by a suitable method such as a pressure-actuated switch operating between suitable limits, such as ¼ lb and 2 lb, by manually or automatically operated switch at definitely timed intervals, or by such other method as is indicated by conditions of the test.

IX. METHOD OF CONDUCTING TESTS

45. The steam pressure during continuous tests shall be maintained at a constant value. In intermittent tests, the steam pressure will vary. Enough readings of the pressure shall be taken at regular intervals in each *on* and *off* period to obtain a true average. A recording pressure gage is strongly recommended for intermittent tests. The water level in the boiler shall be maintained constant during the test.

46. Water Fed or Condensed. The records of water fed or condensed shall be made as required by the measuring system. It is desirable to measure the water fed or condensed each half-hour to check the rate of steaming.

47. Observations of drafts, pressures, and temperatures during *continuous tests* shall be made at regular intervals of not more than 15 min. During *intermittent tests* the flue-gas temperature shall be read a sufficient number of times, at regular intervals, in each *on* and *off* period to obtain a true average. A flue-gas temperature recorder is strongly recommended for intermittent tests. Separator water shall be weighed hourly.

48. Flue gas analyses shall preferably be made on samples collected for periods not exceeding 30 min. In intermittent tests, the gas shall be collected and analyzed separately for the *on* and *off* periods of operation unless a recorder is used. The samples may be held over several successive periods and analyzed hourly.

49. Smoke Readings. When the Ringelmann Chart Method is used, smoke observations shall be made every 30 sec through 1 hour in a continuous test or four complete *on* and *off* cycles in an intermittent test.

50. Attention to Fire. For stokers feeding not more than 60 lb of fuel per hour, intended for automatic operation, no attention shall be given to the fire during test periods other than the necessary removal of clinker or ash. For stokers feeding more than 60 lb of fuel per hour, attention may be given as required for satisfactory operation. The time and nature of each attention shall be recorded; the duration of each cleaning period shall be recorded.

X. PERFORMANCE CHARTS

51. (a) If four or more continuous tests are run at different combustion rates, a performance chart may be made. Against output expressed in thousands of Btu and square feet of steam radiation, the following items may be plotted: stoker and boiler efficiency, flue-gas temperature, combustion rate, unit combustion rate, furnace draft, smokehood draft, windbox pressure, and CO_2. One square foot of steam radiation is defined as 240 Btu per hour.

(b) If four or more intermittent tests are run at varying intervals of operation, a performance chart may be drawn plotting stoker and boiler efficiency against output expressed in thousands of Btu and square feet of steam radiation.

STANDARD FORM

Table 1. Data and Results of Tests

General Information	
1. Number of test	
2. Date of test	
3. Test conducted by	
4. Type of test	
5. Make and catalog designation of stoker	
6. Type of stoker	
7. Maximum feed rate of stoker	
8. Make and catalog designation of boiler	
9. Type of boiler	
10. Commercial rating boiler	sq ft steam radiation
11. Heating surface of boiler	sq ft
12. Furnace volume	cu ft
13. Hearth or grate dimensions, diam..width..length..	ft in.
14. Hearth or grate area	sq ft
15. Weight of water in boiler to normal water line	lb
16. Weight of water in boiler.....full	lb
17. Water space	per cent
18. Steam space	per cent
19. Duration of test	hours
20. Feed rate setting	lb per hour
21. Barometer reading	in. mercury
Fuel	
22. Kind ....Seam....Mine....County....State	
23. Size	
Proximate Analysis, as Fired	
24. Moisture	per cent
25. Volatile matter	per cent
26. Fixed carbon	per cent
27. Ash	per cent
28. Calorific value, as fired	Btu per lb
Ultimate Analysis, as Fired	
29. Carbon	per cent
30. Hydrogen	per cent
31. Oxygen	per cent
32. Nitrogen	per cent
33. Sulphur	per cent
34. Ash	per cent
General Information	
35. Total fuel fired during test period	lb
36. Power consumed by stoker...kwhr per ton of fuel fired	
37. Average interval between attention to fire...hour...min	
38. Average duration of cleaning period	min
Ash and Refuse	
39. Total weight of ash and refuse	lb
40. Combustible in ash and refuse	per cent
41. Carbon burned per pound of fuel, as fired	lb
Drafts and Pressures	
42. Stoker windbox pressure	in. water
43. Draft in furnace	in. water
44. Draft at boiler outlet	in. water
Flue Gases	
45. Carbon dioxide (CO_2)	per cent
46. Oxygen (O_2)	per cent
47. Carbon monoxide (CO)	per cent
48. Nitrogen (N_2) (by difference)	per cent

49. Temperature of flue gases leaving boiler.....F
50. Dry flue gas per pound of fuel, as fired.....lb
51. Excess air .....per cent
52. Temperature of air for combustion, dry-bulb.....F

Steam and Feed Water

53. Steam pressure by gage.....lb per sq in.
54. Absolute steam pressure.....lb per sq in.
55. Temperature of feed water.....F
56. Total weight of feed water.....lb
57. Total weight of water from separator.....lb
58. Moisture in steam.....per cent
59. Total water evaporated.....lb

Hourly Rates

60. Fuel fired (average).....lb per hour
61. Combustion rate (average).....lb per hour
62. Heat release rate (average).....Btu per hour
63. Unit combustion rate (average).....
.....lb per sq ft of hearth or grate area per hour
64. Unit heat release rate (average).....
.....Btu per cu ft furnace volume per hour
65. Actual water evaporated.....lb per hour
66. Correction for uncovered boiler.....lb per hour
67. Total corrected evaporation.....lb per hour

Output

68. Output .....Btu per hour
69. Output .....sq ft steam radiation
70. Heat absorbed by boiler heating surface.....
.....Btu per sq ft per hour
71. Corrected evaporation per pound of fuel, as fired....
.....lb

Heat Balance

	a BTU PER LB FUEL	b PER CENT
72. Heat transferred to boiler (efficiency of boiler and stoker).....	.....	.....
73. Heat carried away by steam in flue gases	.....	.....
74. Heat carried away by dry flue gases	.....	.....
75. Heat lost by not burning carbon monoxide	.....	.....
76. Heat lost by not burning combustible in ash	.....	.....
77. Heat lost due to radiation and unaccounted for	.....	.....
Total	.....	100.0

CALCULATIONS FOR TABLE 1

Wherever CO_2, O_2, CO and N_2 are used they are percentages by volume of these constituents in the gases of combustion.

A_s = The allowed projected area of uncovered metal steam-heated surface.

H = Total heat (in Btu per lb) of dry saturated steam at the boiler outlet pressure.

h = Total heat (in Btu per lb) in feed water at boiler inlet.

t_s = Temperature of steam at boiler outlet pressure.

Formulae and Notes

$$\text{Item 17} = \frac{\text{Item 15}}{\text{Item 16}} \times 100$$

$$\text{Item 18} = \frac{\text{Item 16} - \text{Item 15}}{\text{Item 16}} \times 100$$

Item 36 = Total power consumed during test divided by total fuel fired expressed in tons.

Item 37 = Total time of test divided by number of attention periods.

Item 38 = Sum of duration of cleaning periods divided by the number of cleaning periods.

$$\text{Item 41} = \frac{\text{Item 29}}{100} - \frac{\text{Item 39} \times \text{Item 40}}{\text{Item 35} \times 100}$$

$$\text{Item 50} = \frac{700 + 4CO_2 + O_2}{3\,(CO_2 + CO)} \times \text{Item 41}$$

$$\text{Item 51} = \frac{(O_2 - \tfrac{1}{2}CO)}{0.264N_2 - (O_2 - \tfrac{1}{2}CO)} \times 100$$

Item 54 = (Barometric Pressure in Inches of Mercury × 0.491) + Item 53.

Item 56 = (When condenser is used) weight of water from condenser + Item 57.

$$\text{Item 58} = \frac{\text{Item 57}}{\text{Item 56}} \times 100$$

$$\text{Item 59} = \text{Item 56} - \text{Item 57}$$

$$\text{Item 60} = \frac{\text{Item 35}}{\text{Item 19}}$$

$$\text{Item 61} = \text{Item 60} - \frac{\text{Item 39} \times \text{Item 40} \times 14{,}600}{\text{Item 28} \times 100 \times \text{Item 19}}$$

$$\text{Item 62} = \text{Item 61} \times \text{Item 28}$$

$$\text{Item 63} = \frac{\text{Item 61}}{\text{Item 14}}$$

$$\text{Item 64} = \frac{\text{Item 62}}{\text{Item 12}}$$

$$\text{Item 65} = \frac{\text{Item 59}}{\text{Item 19}}$$

$$\text{Item 66} = \frac{1.8}{H\text{-}h}\,(t_s - \text{Item 52})\,A_s$$

$$\text{Item 67} = \text{Item 65} + \text{Item 66}$$

$$\text{Item 68} = \text{Item 67}\,(H\text{-}h)$$

$$\text{Item 69} = \frac{\text{Item 68}}{240}$$

$$\text{Item 70} = \frac{\text{Item 68}}{11}$$

$$\text{Item 71} = \frac{\text{Item 67}}{\text{Item 60}}$$

$$\text{Item 72}a = \frac{\text{Item 68}}{\text{Item 60}}$$

$$\text{Item 73}a = 9 \times \frac{\text{Item 30}}{100} \times (1090.7 + 0.455 \times \text{Item 49} - \text{Item 52})$$

$$\text{Item 74}a = \text{Item 50} \times 0.24\,(\text{Item 49} - \text{Item 52})$$

$$\text{Item 75}a = \frac{CO}{CO_2 + CO} \times \text{Item 41} \times 10{,}150$$

$$\text{Item 76}a = \left(\frac{\text{Item 29}}{100} - \text{Item 41}\right) \times 14{,}600$$

Item 77a = Item 28 − (Item 72 + Item 73 + Item 74 + Item 75 + Item 76)

$$\text{Items 72}b \text{ to 77}b\text{, inclusive} = \frac{\text{Items 72}a \text{ to 77}a\text{, inclusive}}{\text{Item 28}} \times 100$$

SHORT TEST FORM

Table 2. Data and Results of Tests

General Information

1. Number of test.....
2. Date of test.....
3. Test conducted by.....
4. Type of test.....continuous.....intermittent....
5. Make and catalog designation of stoker.....
6. Type of stoker.....
7. Maximum feed rate of stoker.....
8. Make and catalog designation of boiler.....
9. Type of boiler.....
10. Commercial rating of boiler.....sq ft steam radiation
11. Heating surface of boiler.....sq ft
12. Furnace volume .....cu ft
13. Hearth or grate dimensions, diam..width..length..ft. in.
14. Hearth or grate area.....sq ft
15. Weight of water in boiler to normal water line.....lb
16. Weight of water in boiler.....full.....lb
17. Water space .....per cent
18. Steam space .....per cent
19. Duration of test.....hours
20. Feed rate setting.....lb per hour
21. Barometer reading.....in. mercury

Fuel

22. Kind .....Seam.....Mine.....County.....State
23. Size .....
24. Calorific value, as fired.....Btu per lb
25. Total fuel fired during test period.....lb
26. Average interval between attention to fire.....hour min
27. Average duration of cleaning period.....min

Ash and Refuse

28. Total weight of ash and refuse.....lb
29. Combustible in ash and refuse.....per cent

Drafts and Pressures

	On Periods	Off Periods
30. Stoker windbox pressure.....in. water		
31. Draft in furnace.....in. water		
32. Draft at boiler outlet.....in. water		

Flue Gases

	On Periods	Off Periods
33. Carbon dioxide (CO_2).....per cent		
34. Temperature of flue gases leaving boiler.....F		
35. Temperature of air for combustion, dry-bulb.....F		

Steam and Feed Water

36. Steam pressure by gage.....lb per sq in.
37. Absolute steam pressure.....lb per sq in.
38. Temperature of feed water.....F
39. Total weight of feed water.....lb
40. Total weight of water from separator.....lb
41. Moisture in steam.....per cent
42. Total water evaporated.....lb

Hourly Rates

43. Fuel fired (average).....lb per hour
44. Combustion rate (average).....lb per hour
45. Heat release rate (average).....Btu per hour
46. Unit combustion rate (average).....
.....lb per sq ft of hearth or grate area per hour
47. Unit heat release rate (average).....
.....Btu per cu ft furnace volume per hour
48. Actual water evaporated.....lb per hour
49. Correction for uncovered boiler.....lb per hour
50. Total corrected evaporation .....lb per hour

Output

51. Output .....Btu per hour
52. Output .....sq ft steam radiation
53. Heat absorbed by boiler heating surface.....
.....Btu per sq ft per hour
54. Corrected evaporation per pound of fuel as fired.....lb

Efficiency

55. Heat transferred to boiler (efficiency of boiler and stoker).....per cent

CALCULATIONS FOR TABLE 2

Wherever CO_2, O_2, CO and N_2 are used they are percentages by volume of these constituents in the gases of combustion.

A_s = The allowed projected area of uncovered metal steam-heated surface.

H = Total heat (in Btu per lb) of dry saturated steam at the boiler outlet pressure.

h = Total heat (in Btu per lb) in feed water at boiler inlet.

t_s = Temperature of steam at boiler outlet pressure.

Formulae and Notes

$$\text{Item 17} = \frac{\text{Item 15}}{\text{Item 16}} \times 100$$

$$\text{Item 18} = \frac{\text{Item 16} - \text{Item 15}}{\text{Item 16}} \times 100$$

Item 26 = Total time of test divided by number of attention periods.

Item 27 = Sum or duration of cleaning periods divided by the number of cleaning periods.

Item 37 = (Barometric Pressure in Inches of Mercury × 0.491) + Item 36.

Item 39 = (When condenser is used) weight of water from condenser + Item 40.

$$\text{Item 41} = \frac{\text{Item 40}}{\text{Item 39}} \times 100$$

Item 42 = Item 39 — Item 40

$$\text{Item 43} = \frac{\text{Item 25}}{\text{Item 19}}$$

$$\text{Item 44} = \text{Item 43} - \frac{\text{Item 28} \times \text{Item 29} \times 14{,}600}{\text{Item 24} \times 100 \times \text{Item 19}}$$

Item 45 = Item 44 × Item 24

$$\text{Item 46} = \frac{\text{Item 44}}{\text{Item 14}}$$

$$\text{Item 47} = \frac{\text{Item 45}}{\text{Item 12}}$$

$$\text{Item 48} = \frac{\text{Item 42}}{\text{Item 19}}$$

$$\text{Item 49} = \frac{1.8}{H\text{-}h} (t_s - \text{Item 35}) A_s$$

Item 50 = Item 48 + Item 49

Item 51 = Item 50 (H-h)

$$\text{Item 52} = \frac{\text{Item 51}}{240}$$

$$\text{Item 53} = \frac{\text{Item 51}}{\text{Item 11}}$$

$$\text{Item 54} = \frac{\text{Item 50}}{\text{Item 43}}$$

$$\text{Item 55} = \frac{\text{Item 51} \times 100}{\text{Item 43} \times \text{Item 24}}$$

The Flow of Air Through Exhaust Grilles

By A. M. Greene, Jr.* and M. H. Dean** (*MEMBER*), Princeton, N. J.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Princeton University.

THE primary object of this research problem was the determination of a coefficient or series of coefficients to be used in calculating the air flow through an exhaust grille of modern form from anemometer readings.

It will be recalled that this subject was treated by Davies in papers[1] previously presented before the Society in which he proposed the following formula for exhaust grilles:

$$Cfm = K\,V\,A \qquad (1)$$

in which:

V = Corrected velocity from anemometer in feet per minute.
A = Gross area in square feet (duct size).
K = An average coefficient dependent on velocity.

For K, Professor Davies found average values ranging from 0.762 at low velocities to 0.832 at high velocities, the mean of these being 0.80.

With the increase in the per cent free area of grilles in recent years, the applicability of this factor for grilles of more modern design has been questioned. This paper reports tests to determine the applicability of the earlier formula to recent designs of grilles.

The results of this research indicate that a simple general formula for air discharge to be used with exhaust grilles of recent design within a maximum error of ±6 per cent is

$$Cfm = 0.873\,V\,A \qquad (2)$$

where:

V = Velocity in feet per minute indicated by corrected anemometer readings.
= Anemometer reading for complete moving traverse with automatic weighted value divided by total time and corrected by anemometer calibration.
A = Gross frontal area = the smallest area of the inside of the frame of the grille in square feet.

The effects of size of grille and velocity with a smaller possible maximum error are eliminated by the use of a coefficient K in place of the average coefficient 0.873. K is determined by the following formula:

$$K = \frac{0.904}{R^{0.04}}\left(1 + \frac{V - 600}{40{,}000}\right) \qquad (3)$$

where:

K = Coefficient in the formula, $Cfm = K\,V\,A$
R = Hydraulic radius of duct in inches =

$$\frac{\text{Nominal Area in square inches}}{\text{Nominal Perimeter in inches}}$$

V = Velocity in feet per minute as indicated by anemometer.

The word *nominal* refers to the dimensions of the duct into which the grille under consideration fits.

The results of this research indicate the necessity of making measurements in a definite manner and with definite standard equipment and also the desirability of formulating a code for the determination of the air discharge through exhaust grilles.

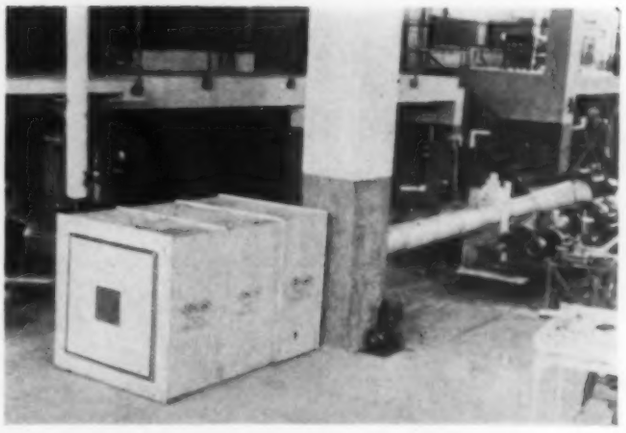

Fig. 1—Arrangement of apparatus

Design of Test Equipment

The following five points were considered in the design of the testing equipment for exhaust grilles:

1. A simple method must be employed to measure the true quantity of air passing through the system.
2. There must be an easy method of changing the size and type of grille.
3. An air-tight system is imperative.
4. The air volume must be controlled.
5. The air must not be in an eddy condition upon entering the measuring device.

Keeping these points in mind the equipment was designed as shown in Figs. 1 and 2.

The duct of 12 in. inside diameter was connected to a 4 ft x 4 ft x 6 ft plenum chamber. This chamber acted as a stilling box for the air since the cross-sectional area was very large in comparison to the free area of the grille. It also served as an excellent device for mounting the grilles, which were set in pieces of plywood of a size to fill the opening in the end of the plenum chamber. The plywood was mounted on the plenum chamber by

*Dean and Professor of Mechanical Engineering, School of Engineering, Princeton University.
**Dean-Hagny Corp., Kansas City, Mo., formerly Graduate Student, Princeton University.

[1] "The Measurement of the Flow of Air Through Registers and Grilles," by L. E. Davies. (A. S. H. V. E. TRANSACTIONS: Vol. 36, 1930, p. 201; Vol. 37, 1931, p. 619; Vol. 39, 1933, p. 373.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

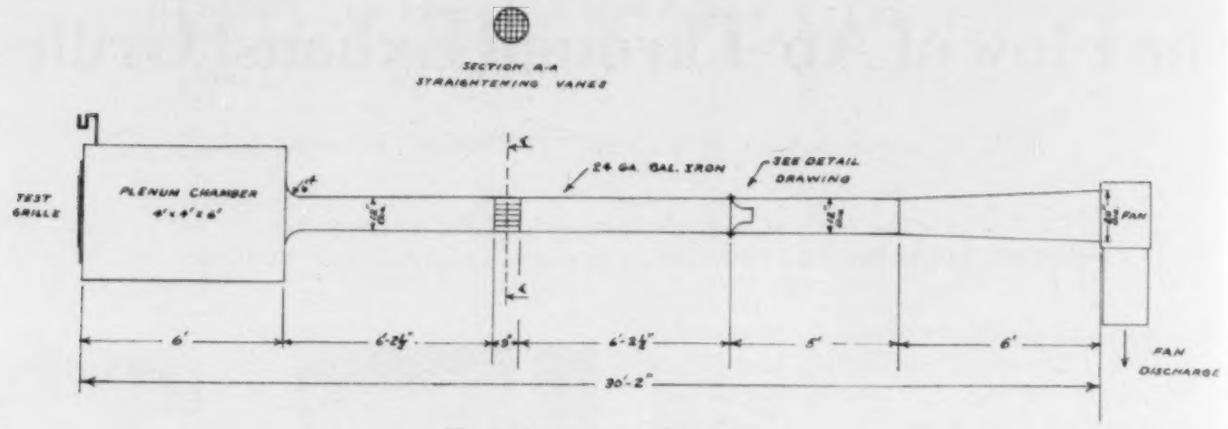

Fig. 2—Arrangement of apparatus

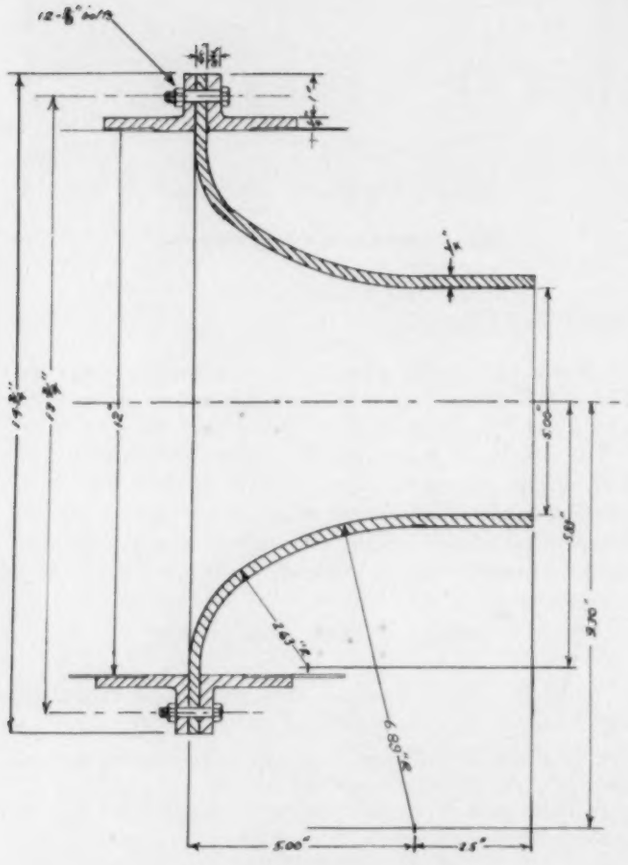

Fig. 3—Section through nozzle

means of bolts through the sheet metal and backed by a metal frame of angle irons. Soft putty was filled in around the board to prevent any possible air leakage. All other joints in the system were soldered.

The fan was driven by a variable speed motor and the fan discharge area was variable.

It will be noted in Fig. 2 that the air left the plenum chamber through a large sheet metal nozzle of throat diameter equal to the duct diameter. This prevents irregular flow due to the contraction of air passage from the area of the plenum chamber to that of the duct. Straightening vanes of the *egg crate* type were made of 2 in. squares 9 in. deep.

The nozzle for the measurement of the flow of air, shown in Figs. 3 and 4, was of the elliptical profile type. It was made of dimensions identical to those of the *U. S. Bureau of Standards* Nozzle C-2.[2]

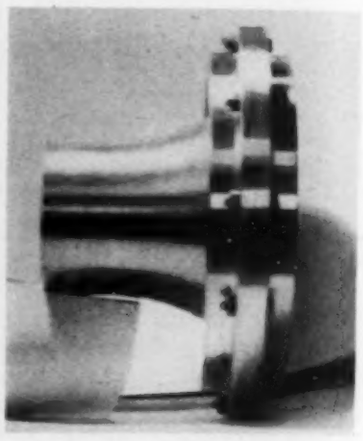

Fig. 4—Nozzle

The Nozzle and Its Calibration

The authors were somewhat at a loss to choose the best method of measuring the quantity of air. A water displacement tank was suggested and discarded because of cost and difficulty in handling. The choice then remained between a venturi meter, a nozzle, a flat plate orifice, and a Pitot tube. Of these types the nozzle was chosen because of its accuracy and the simplicity of mounting it in the 12 in. duct. Although many recent papers have dwelt upon the value of the flat plate orifice, its coefficients have greater limits of variation than those of the nozzle or the venturi meter. The Pitot tube, successfully employed by Davies, would have been an ideal method of measuring the air but its use was dispensed with since 20 grilles were to be tested at various air velocities and duct Pitot tube traverses and anemometer grille traverses could not be taken at the same time by a single observer. Therefore, it was decided to use a metering device which gave the quantity of air flow from a single reading.

In order to have a check on the nozzle coefficient, a

[2] *U. S. Bureau of Standards, Journal of Research*, Vol. 2, 1929, No. 3, p. 561.

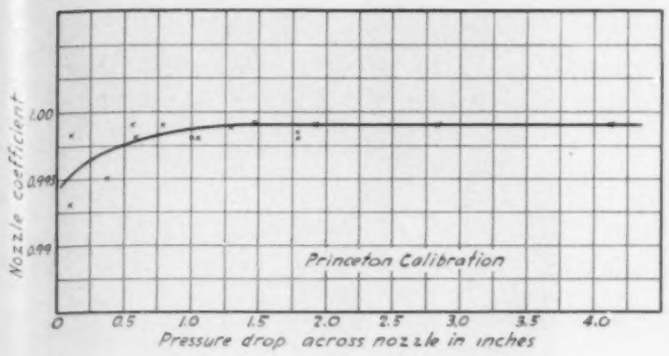

Fig. 5—Coefficients for nozzle

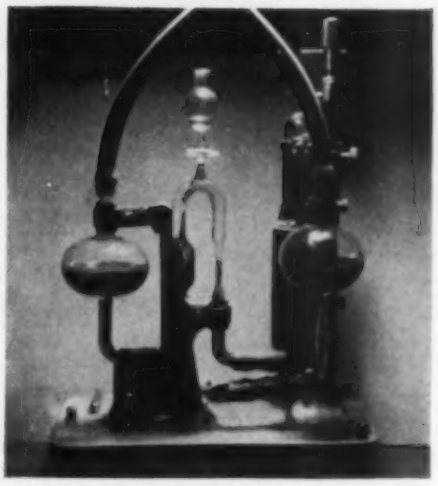

Fig. 7—Wahlen gage

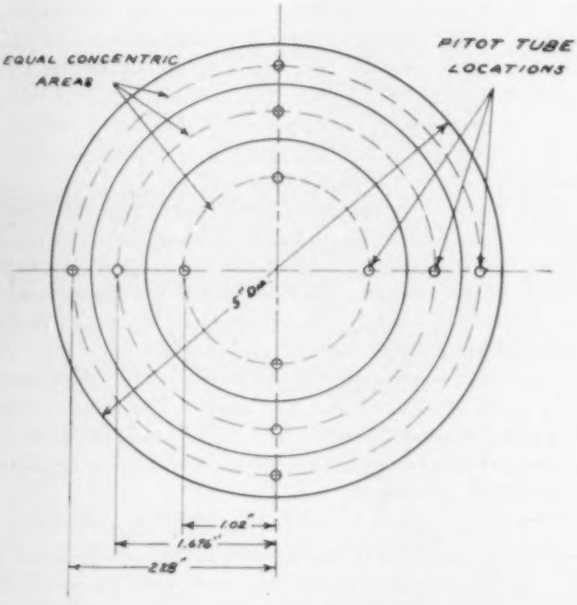

Fig. 6—Locations of Pitot tube at nozzle throat

nozzle which had been built and calibrated was followed in design. As stated before, this nozzle was the type C-2 of the *U. S. Bureau of Standards* Nozzle for which the coefficient varies from 0.993 to 0.995.[3] The results of the calibration of the nozzle used in the tests described in this paper are seen in Fig. 5 to vary from 0.994 to 0.999, which gives a maximum difference of four-tenths of one per cent from the results obtained by the Bureau of Standards. The pressure drop was measured through 1/16 in. flush holes at the plane of the outlet to the nozzle and at 17½ in. upstream from this point.

The calibration was performed by means of a Standard A.S.H.V.E. Pitot tube using a coefficient of unity as should be employed where the air at the nozzle throat or point of Pitot tube traverse is in streamline flow.

Four Pitot tube readings were taken at quartering points in the mid-area points of three imaginary concentric sections of equal areas formed in the 5 in. throat diameter. The location of these readings is shown in Fig. 6.

In using the Pitot tube, the velocity determined from the gage reading at any point, x, applies to an element of cross-sectional area of $2\pi r_x dr$ so that the volume flowing per second through a circular area is given by

$$\text{Volume per Second} = 2\pi \int r_x V_x dr = \pi \int V_x d(r_x)^2 \quad \ldots\ldots (4)$$

It is seen from this equation that if the velocity is plotted as ordinate and the square of the radius as abscissa, the area under the curve when multiplied by π will give the discharge per second. Since the partial areas to which each set of Pitot tube readings refer are equal, this plotting is unnecessary and the average velocity for each set may be averaged to obtain the mean velocity through the nozzle. The velocity in feet per minute from the Pitot tube reading, h, in inches of water, and with air of relative density, x, is given by Equation 5:[4]

$$V = 4006\sqrt{h/x} \quad \ldots\ldots (5)$$

The four readings of h of the Pitot tube in any zone were averaged, and the mean value of $\sqrt{h}$ for the full nozzle was found by extracting the square root of the average reading for each zone and then determining the average of the square roots of the average of the zone readings.

Since the Pitot tube readings apply to equal annular areas, the average $\sqrt{h}$ can be applied to the full area, and the true quantity of air delivered from the nozzle can be found by multiplying the average velocity by the outlet area of the nozzle.

The quantity of air at outlet flowing through the nozzle as indicated by the pressure drop is given by formula 6:[5]

$$\text{Volume per Second} = 8.02 A_2 \sqrt{\frac{\Delta p \times v_1}{1 - \left(\frac{A_2}{A_1}\right)^2 \left(1 - 1.428\,\frac{\Delta p}{p_1}\right)}} \quad \ldots\ldots (6)$$

[3] Loc. Cit. See Note 2, p. 652.

[4] See Appendix No. 1.
[5] See Appendix No. 2.

in which:

A_2 = Nozzle throat area in square feet.

A_1 = Duct area in square feet.

ΔP = Pressure drop from point p_1 above nozzle to point p_2 following nozzle in pounds per square feet.

v = Volume of 1 lb of air under conditions of test.

Having solved for the quantity as determined by the Pitot tube and for that indicated by the pressure drop across the nozzle, the nozzle coefficient was determined at various velocities by dividing the Pitot tube quantity by the nozzle quantity. The result of this calibration is shown in Fig. 5.

The results of Pitot tube traverses demonstrate that the Pitot tube does have a noticeable interference with air flow. Although the velocity pressure is for the most part correct at a maximum in the center of the nozzle, the pressure drop across the nozzle, which should remain constant, has a gradual decrease as the Pitot tube is passed in front of the outlet. This is explained by the fact that the tube, upon insertion into the duct, sets up eddies and friction losses, and, therefore, the fan draws less air through the nozzle. It will be apparent that these values are not large enough to affect appreciably the accuracy of the work, yet they do exist and are here recorded with the idea of stimulating some additional work on the properties of Pitot tubes and the feasibility of using streamlined tubes.

In order to eliminate errors due to different gage characteristics, a single gage was used on any two sets of comparative readings. All readings below an inch of water were taken on a Wahlen gage, whereas larger readings were taken on an inclined draft gage. These gages were also checked against one another and found to give identical results.

The density of the alcohol in the Wahlen gage was 0.8142, while the kerosene and ligroin mixture in the inverted U-tube had a density of 0.8048. The difference between these two densities is 0.0094, which is ideal for the operation of the gage. These densities were determined by means of the displacement method using gravimetric scales. The readings on the Wahlen gage were changed from inches of alcohol to inches of water by multiplying the gage reading by the density of the alcohol (0.8142).

Method of Anemometer Traverse

Before making an attempt to determine the discharge coefficients for grilles, it was necessary to experiment with the various methods of taking a traverse with an anemometer in contact with the grille. It was immediately discovered that the readings near the center of all types of grilles were much higher than those near the edges and it occurred to the observer that perhaps some allowance should be made for this variation.

A series of observations also showed that there was an appreciable effect due to the obstruction of the hand in holding the anemometer. To overcome this, a ⅜ in. hollow rod 2 ft long was attached to the top of the anemometer. Through the hole in the rod a stiff wire was connected to the starting and stopping catch on the anemometer, so that it was controlled at the end of the rod. Time was taken by an electric timer which rang a bell every 15 seconds, and, therefore, complete attention

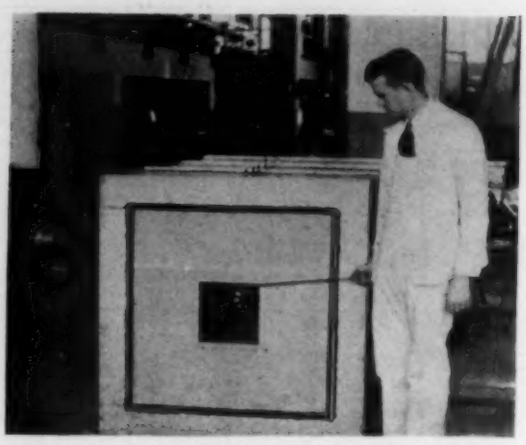

Fig. 8—Application of anemometer

could be given to the making of the traverse. All tests were made using the electric timer and the rod-mounted anemometers, although several check runs were made using a stop watch. In making a continuous type traverse, it was found that 15 seconds was a convenient time for making one crossing of both the 1 and 2 ft grilles. No advantage was gained by spending 30 seconds on the larger size.

Both a 3 in. and a 4 in. anemometer were used in all but the first few tests, and it was found that the instruments checked within two or three per cent of each other. The 4 in. anemometer was calibrated for these tests by the Bureau of Standards and the new 3 in. instrument was calibrated by the manufacturer before delivery. The method of holding the instrument while taking a traverse on a small grille is shown in Fig. 8. The authors found that the 3 in. anemometer was more easily handled and is more applicable to the method of traverse to be recommended by this paper.

The following methods of traverse were employed:

1. *Spot traverse with straight average.* In this method the 12x12 grilles were divided into nine equal zones, and a half minute reading was recorded for each zone. A straight average was obtained which was obviously too low, due to the fact that too much value was given the readings along the edges.

2. *Continuous traverse across the grille.* In this method the anemometer is moved back and forth across the grille without repeating on any strip. This method gives approximately the same results as the previous one. Methods 1 and 2 were recommended by Davies.[6]

3. *Zone traverse with weighted average.* The grille was divided into three horizontal zones and a moving traverse was made in each zone for a period of 1 min. In averaging these three readings the middle zone is given four times the value of the top and bottom zones. This is in accordance with Simpson's Rule in which the mean height of a parabolic curve may be determined by dividing the length into two equal parts, and then measuring the height of the three boundary lines of these parts. The mean height of the curve is the weighted average of four times the center line plus the height of each of the other two outer lines.

4. *Spot traverse with weighted average.* In this method the values of the nine individual readings were weighted according to Simpson's Rule. In this case the center reading is multiplied by sixteen, each of the intermediate side values by four, and the corner values by one. Although this method is more nearly

[6]The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies. (A. S. H. V. E. Transactions, Vol. 39, 1933, p. 382.)

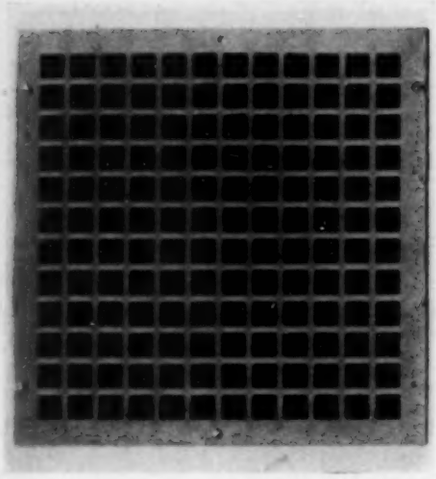

Fig. 9—Type 1 grille 58.7 per cent free area

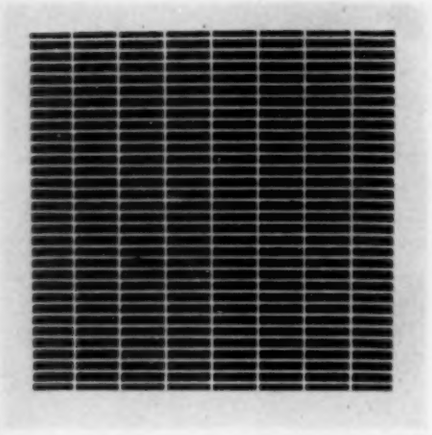

Fig. 10—Type 2 grille 74.98 per cent free area

correct, it is too involved and time consuming to be of practical value.

5. *Moving traverse with automatic weighted value.* The grille was divided into three zones as in Method 3, and a continuous traverse was made with one crossing on the upper zone, four crossings on the center zone, and one crossing on the lower zone. This method automatically applied Simpson's Rule and gave the same results as methods three and four.

The authors applied the fifth method to all grilles and all sizes and found that it gave a curve of coefficients slightly more uniform than that from the straight average method. In all of these the anemometer was moved in contact with the face of the grille.

The Exhaust Grille Coefficient *K*

Grille traverses were made in accordance with Method 5, which is termed: *Moving traverse with automatic weighted value.* Two or more runs were taken with each anemometer at each velocity, and the values were corrected by the calibration curves and averaged. The indicated volume of air passing the grille is, of course, obtained in accordance with the Equation of Continuity of Flow by multiplying the indicated velocity by an area, and, if necessary, a discharge coefficient. One of the major problems of this paper was to determine which grille area should be used, and, having chosen a particular area, to find out how it applied to grilles of various sizes and various per cents of free area.

The type grilles used in this investigation are shown in Figs. 9, 10, 11, 12 and 13. These five types of grilles were tested in the following four sizes: 24x6; 12x12; 24x12; 24x24.

Fig. 11—Type 3 grille 73.36 per cent free area

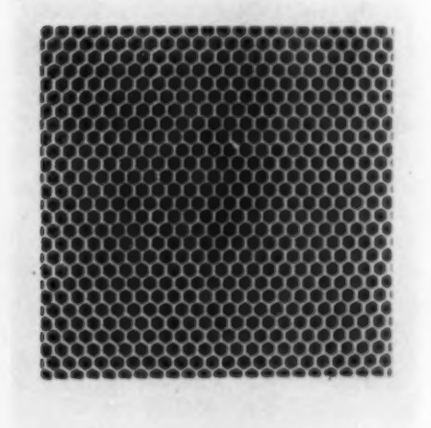

Fig. 12—Type 4 grille 90.5 per cent free area

Fig. 13—Type 5 grille 77.6 per cent free area

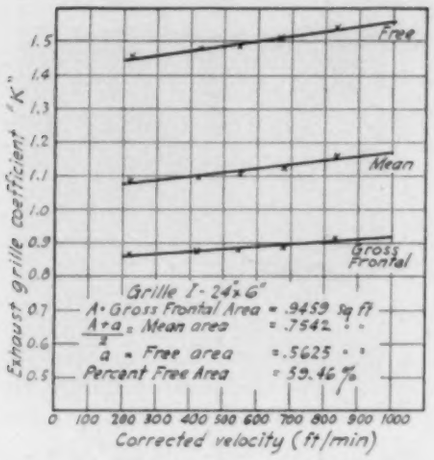
Exhaust grille coefficient "K"
Free
Mean
Gross Frontal
Grille 1-24"x6"
A = Gross Frontal Area = .9459 Sq ft
(A+a)/2 = Mean area = .7542 " "
a = Free area = .5625 " "
Percent Free Area = 59.46 %
Corrected velocity (ft/min)

Fig. 14—Coefficients on gross and free areas

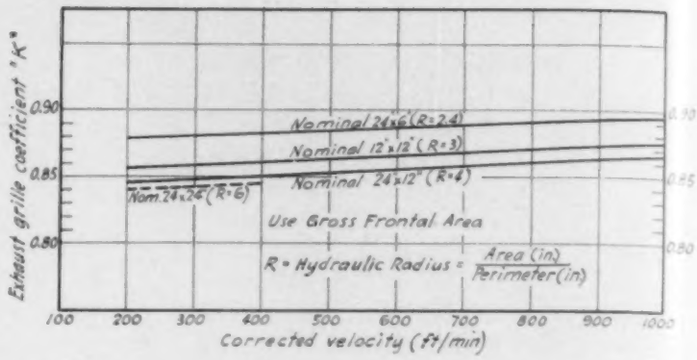
Exhaust grille coefficient "K"
Nominal 24x6 (R=2.4)
Nominal 12"x12" (R=3)
Nominal 24"x12" (R=4)
Nom 24"x24" (R=6)
Use Gross Frontal Area
R = Hydraulic Radius = Area (in) / Perimeter (in)
Corrected velocity (ft/min)

Fig. 15—Curves showing variation of mean coefficient with grille size

This paper summarizes the results of tests on 20 grilles at anemometer velocities varying from 200 fpm to 1000 fpm. It will be noted in the curves that follow that the velocity range was limited on the largest size grille due to capacity limitations of the fan on account of the excessive drop in pressure at the nozzle.

The discharge coefficients of these grilles were found by dividing the true flow from the calibrated nozzle by the volume indicated by the product of the anemometer reading and an area. The following areas were considered as possibilities:

1. The Free Area, which is defined as the open area of the grille through which the air actually passes.
2. The Gross Frontal Area is the smallest area of the inside of the frame of the grille. This is the area which the anemometer crosses in making the traverse.
3. The Mean Area is the average of the Free Area and the Gross Frontal Area.
4. The Nominal Area (called Gross Area by Davies) is the area of the duct.

The per cent free areas as given in this paper are determined by the ratio:

$$\text{Per cent Free Area} = \frac{100 \times \text{Free Area}}{\text{Gross Frontal Area.}}$$

All these values were computed by the authors and compared with those furnished by the manufacturers, although the authors' values were used in all cases.

Upon studying the 20 curves of coefficients plotted against velocity, the following observations are made:

1. The coefficient as determined from the gross frontal area, as in Fig. 14, is relatively constant over the complete range of grilles including all types and all sizes, whereas the coefficients using the mean and free areas vary inversely as the increasing per cent free area. This definitely shows that the coefficient should not be based on the mean or free areas, unless the lower coefficient is to be used to illustrate the advantages of the grilles of higher per cent free area.
2. The curves, such as shown in Fig. 15, demonstrate the effect of size of grille and show that the smaller grilles have a slightly higher coefficient. A study of these curves and the hydraulic radii of the duct was used in developing the empirical formula to be given later.
3. Figs. 16 and 17 show a striking proof of the value of using the gross frontal area instead of the previously used nominal area. The gross frontal areas for the various types of grilles are given:

Grille No. 1 Gross Frontal of 12 × 12 = 0.9604 sq ft.
Grille No. 2 Gross Frontal of 12 × 12 = 0.9538 sq ft.
Grille No. 3 Gross Frontal of 12 × 12 = 0.8633 sq ft.
Grille No. 4 Gross Frontal of 12 × 12 = 0.8556 sq ft.
Grille No. 5 Gross Frontal of 12 × 12 = 0.8636 sq ft.

By examining the points on Fig. 16, which gives the exhaust grille coefficients based on the nominal or duct area, it is seen that Grilles 1 and 2, which have much larger gross frontal areas, lie above the average dotted line, whereas the remaining grilles lie below the line. Hence, by using the gross frontal area, the values are brought nearer to a straight line as shown in Fig. 17. The possible per cent of error is practically halved by using the curve as determined by the gross frontal area, because an allowance for the design variations of the various manufacturers has been made.

It is then proposed that the following equation and coefficients be used:

$$Cfm = K\ V\ A \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (7)$$

where

K = Values as given in the tabulation of the next paragraph.
V = Velocity in feet per minute as indicated by a calibrated anemometer using Method 5.
A = Gross frontal area in square feet.

By using a value of $K = 0.873$ for all grilles and sizes, the maximum observed error was ± 6.07 per cent, and the average error was ± 3 per cent.

The maximum error resulting from the use of a simpler coefficient was reduced to ± 5.83 per cent, by the use of coefficients varying with velocities. These values of K for different velocities were found from the curves and are given:

Velocity fpm	Coefficient
200	0.863
300	0.865
400	0.867
500	0.871
600	0.872
700	0.875
800	0.877
900	0.880
1000	0.882

By using the accompanying coefficients for grilles of different sizes, the maximum recorded error was ± 4.62 per cent. These values were taken from diagrams of the investigation, using 600 fpm as the average velocity.

K of Nominal 24 × 6 grille = 0.886
K of Nominal 12 × 12 grille = 0.865
K of Nominal 24 × 12 grille = 0.855
K of Nominal 24 × 24 grille = 0.842 (Calculated)

The velocity effect on the given values of K may be cared for by multiplying each of the factors by

$$1+\frac{V-600}{40{,}000} \qquad (8)$$

and the resulting value of K showed a maximum error of ± 4 per cent.

For grilles of all sizes including those given, the effects of size and velocity may be included by using the formula in determining the value of K:

$$K=\frac{0.904}{R^{0.04}}\left(1+\frac{V-600}{40{,}000}\right) \qquad (9)$$

in which

K = Coefficient in formula $Cfm = K\,V\,A$

R = Hydraulic Radius = $\dfrac{\text{Nominal Area in square inches}}{\text{Nominal Perimeter in inches}}$

V = Velocity in feet per minute as indicated by the corrected anemometer using Method 5.

The maximum recorded error in using this formula was ± 5.65 per cent and the average was ± 3.22 per cent.

This remaining error is due to the variation in anemometer readings and to the frictional effect of the grille members. This latter error can only be eliminated by using the test grille coefficients which are recorded by most manufacturers for each of their grilles.

Conclusions

This investigation has checked the work of Davies and established an improved method of determining the quantity of air flowing through a modern exhaust grille. It has been demonstrated that a relatively constant coefficient of $K = 0.873$ is obtained when the gross frontal area is used in the formula:

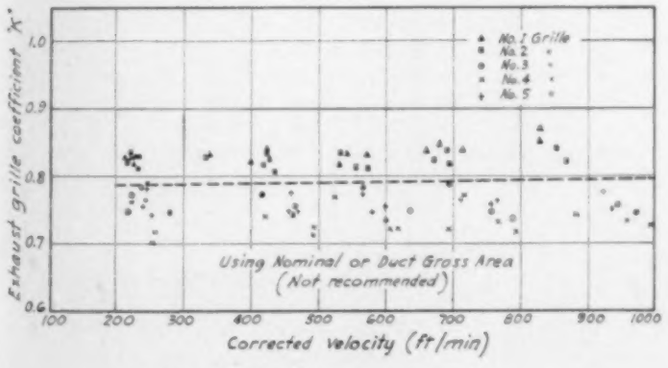

Fig. 16—Coefficients on duct area

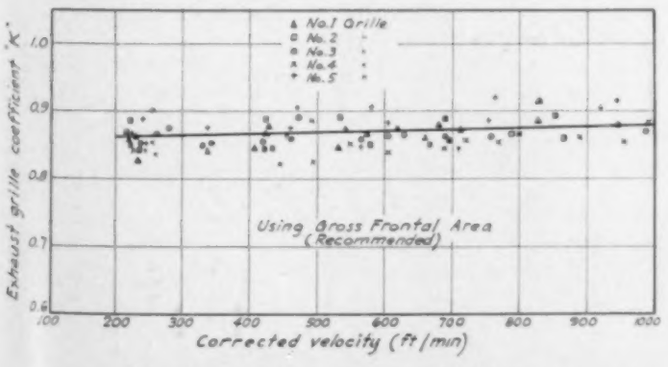

Fig. 17—Coefficients on gross frontal area

$$Cfm = K\,V\,A \qquad (10)$$

The average coefficient for all grilles and all sizes has been changed from the Davies' value of 0.80 to 0.873. These results were obtained by Method 5, *moving traverse with automatic weighted value*, as proposed in this paper.

The variation in K, the maximum recorded error of which was 6 per cent on each side of the mean curve, is dependent upon velocity, grille size, and design. The first two of these variables were cared for in the empirical formula derived from the observed data, whereas the authors were unable to find a law correcting for grille design. The empirical formula is:

$$K=\frac{0.904}{R^{0.04}}\left(1+\frac{V-600}{40{,}000}\right) \qquad (11)$$

When greater accuracy is desired, it is necessary to use coefficients determined for the particular type of grille form and hydraulic radius of duct.

Appendix No. 1

Equation 5 for finding the velocity from the Pitot velocity head, $V = 4006\sqrt{h/x}$ is derived as follows:

$$v^2 = 2gh$$
$$v = \sqrt{2gh}$$

Volume of 1 lb of dry air at 70 F = 13.35 cu ft.
Weight of 1 cu ft water at 70 F = 62.295 lb.

$$\text{Velocity in feet per minute} = 60\sqrt{\frac{64.4\times\dfrac{62.295\times 13.35}{12}\times h}{x}}$$

$$\text{Velocity in feet per minute} = 4005.9\sqrt{h/x} = 4006\sqrt{h/x}$$

in which:

V = Velocity in feet per minute.
h = Velocity head in inches of water at 70 F.
x = Density of actual air relative to air at 70 F and 29.92 in. Hg.

Appendix No. 2

Equation 6 in this paper is derived as follows: From thermodynamics and the principle of conservation of energy, the equation for the gain in kinetic energy of a perfect gas from the change in enthalpy (equation of flow) is

$$\frac{V_2^2-V_1^2}{2g}=\frac{kp_1v_1}{k-1}-\frac{kp_2v_2}{k-1} \qquad (1)$$

this gives

$$\frac{V_2^2}{2g}\left[1-\left(\frac{V_1}{V_2}\right)^2\right]=\frac{k}{k-1}(p_1v_1-p_2v_2)$$

and

$$\frac{V_2^2}{2g}=\frac{\dfrac{k}{k-1}p_1v_1\left(1-\dfrac{p_2v_2}{p_1v_1}\right)}{1-\left(\dfrac{V_1}{V_2}\right)^2} \qquad (2)$$

but

$$\frac{p_2v_2}{p_1v_1}=\frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{\frac{1}{k}}=\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \quad \text{.....(3)}$$

and

$$V_1=\frac{Vol_1}{A_1}=\frac{MRT_1}{p_1A_1} \qquad V_2=\frac{MRT_2}{p_2A_2}$$

hence

$$\frac{V_1}{V_2}=\frac{T_1A_2p_2}{T_2A_1p_1}$$

but

$$\frac{T_1}{T_2}=\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}$$

and therefore

$$\frac{V_1}{V_2}=\frac{A_2}{A_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{k}} \quad \text{.....(4)}$$

substituting equations (3) and (4) in equation (2):

$$\frac{V_2^2}{2g}=\frac{\frac{k}{k-1}p_1v_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}$$

but

$$p_1v_1=RT_1$$

hence

$$\frac{V_2^2}{2g}=\frac{\frac{k}{k-1}RT_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}}$$

and

$$\text{Volume per Second} = A_2V_2 = A_2\sqrt{2g\frac{k}{k-1}R}\sqrt{\frac{T_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{2/k}}}$$

$$\text{Volume per Second} = A_2\sqrt{64.4\frac{1.4}{0.4}\times 53.34}\sqrt{\frac{T_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{2/k}}}$$

$$=110A_2\sqrt{\frac{T_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{p_2}{p_1}\right)^{2/k}}} \quad \text{.....(5)}$$

In order to simplify

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}=\left(\frac{p_1-\Delta p}{p_1}\right)^{\frac{0.4}{1.4}}=\left(1-\frac{\Delta p}{p_1}\right)^{\frac{0.4}{1.4}}$$

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}=1-\frac{0.4}{1.4}\frac{\Delta p}{p_1}$$

(since $\frac{\Delta p}{p_1}$ is of the order of $\frac{1}{50}$ the higher powers are neglected)

Similarly

$$\left(\frac{p_2}{p_1}\right)^{2/k}=1-\frac{2}{1.4}\frac{\Delta p}{p_1}$$

Substituting in equation (6)

$$\text{Volume per second} = 110A_2\sqrt{\frac{T_1\left[1-\left(1-\frac{0.4}{1.4}\frac{\Delta p}{p_1}\right)\right]}{1-\left(\frac{A_2}{A_1}\right)^2\left(1-\frac{2}{1.4}\frac{\Delta p}{p_1}\right)}}$$

$$\text{Volume per second} = 110A_2\sqrt{\frac{\frac{T_1}{p_1}\left(\frac{0.4}{1.4}\Delta p\right)}{1-\left(\frac{A_2}{A_1}\right)^2\left(1-\frac{2\Delta p}{1.4}\right)}}$$

but

$$\frac{T_1}{p_1}=\frac{v_1}{R}$$

$$\therefore \text{Volume per second} = 110A_2\sqrt{\frac{\frac{0.4}{1.4}\times\frac{1}{53.34}\times\Delta p\,v_1}{1-\left(\frac{A_2}{A_1}\right)^2\left(1-1.4286\frac{\Delta p}{p_1}\right)}}$$

or

$$\text{Volume per second} = 8.02A_2\sqrt{\frac{\Delta p\times v_1}{1-\left(\frac{A_2}{A_1}\right)^2\left(1-1.428\frac{\Delta p}{p_1}\right)}}$$

Note: This is volume at discharge. The volume at entrance is equal to the above multiplied by the factor $\left(1-0.714\frac{\Delta p}{p_1}\right)$

CORRELATING THERMAL RESEARCH

As a part of the efforts of the A.S.H.V.E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A.S.H.V.E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.
F. C. Houghten, *Director*.

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

Architecture in Glass. *Architectural Forum*, Feb. 1937, pp. 82-86. Photos of glass walls, floors and furniture in England are shown. Panels, ceilings, walls and doors in France and gilded and decorated glass on the Normandie are also illustrated.

•

Photos of houses and interiors using glass-block walls, ibid Feb. 1937, page 108.

•

Glass at Paris International Exhibition. *Deut. Glaserztg.* vol. 48, page 351, 1937. The glass house erected by St. Gobain is described. Brown porous glass brick having only one-tenth the weight of ordinary glass brick are used in the interior.

•

Glass the Building Material of the 20th Century, by K. K. Perslee. *Tcheco-Verre*, vol. 4, page 99-101 and 115-117, 1937. Discussion of the increasing use of glass in decorative forms for domestic, public, and works buildings.

•

Glass for Constructional Purposes, by J. Polivka. *Journal Society of Glass Tech.*, vol. 20 no. 82, page 764-768, 1936.

•

Heat-Absorbing Glass, by V. Ctyroky and M. Fanderlik. *Sklarske Rozhledy* vol. 14, page 69-73, 1937. Test results on the heat-absorbing glasses, Dethermalglas L and N made by Bilinskasklarna Engels and Co. With unfavorable conditions (wind normally giving adequate cooling), the greatest amount of heat which could penetrate into a room was 86.0 for ordinary window glass, 59.7 for Dethermalglas L, and 52.6% for Dethermalglas N. Better results were obtained by double glazing, the outer window being of Dethermalglas and the inner of ordinary window glass. Double window glazing with ordinary window glass let through a maximum of 76.7% of the incident radiation; with Dethermalglas L on the outside this was reduced 41.3%, and with Dethermalglas N it was reduced to 32.1%. Dethermalglas L was only faintly green, while Dethermalglas N was more deeply colored.

•

Panel Heating. *Domestic Engineering*, vol. 152, no. 2, Aug. 1938, page 48-51, 8 illustrations. Discussion of panel heating. Construction details of a modern home panel heated. The owner states that panel heating functioned perfectly giving the general effect of comfort sensations produced by a sunny spring day.

•

A Low Cost All-Steel House. *American Artisan*, vol. 107, no. 7. July 1938, page 36-37. Photo of house. Construction details. Description of the first Blaw-Knox All Steel experimental home. The design of wall and roof construction has been made to give a maximum of insulation to provide homes warm and easy to heat in winter and cool and comfortable in summer. The building insulation rating is about .10 compared to conventional frame or brick veneer construction of about .25.

•

Nomograph Method of Humidity Control, by R. G. Bateson. *Engineering*, vol. CXLVI, July 8, 1938, page 57. Description of a mechanical humidity nomogram and humidity control instrument.

•

Graphic Solutions—Air and Water Vapor. *Domestic Engineering*, vol. 152, no. 2, Aug. 1938, page 60-61. The properties of air and water vapor are discussed, which cover some of the fundamentals of air conditioning.

•

Equation of State for Steam, by Dr. Jan Juza. *Engineering*, vol. CXLVI, July 1, 1938, page 1-3 and ibid July 8, 1938, page 34-36. Czechoslovakia, Germany, Great Britain and the U. S. are cooperating to produce uniform and internationally acceptable steam tables. This paper deals with results of steam research in Czechoslovakia, obtained by the Masaryk Academy of Work (M. A. P.). Great progress is shown in the Keenan-Keyes Steam Tables and the Kock Steam Tables of 1936-1937 over the 1932 tables.

•

Newspaper Puts Exhaust to Work. *Power*, vol. 82, no. 8, Aug. 1938, page 50-51. Description of newspaper power plant including exhaust steam system used for cooling and dehumidifying, heating and humidifying, conditioning systems. This is said to be the first completely air conditioned newspaper building.

•

Fuel of the Future, by Jos. W. Hays. *Power*, vol. 82, no. 7, July 1938, page 57-58. Discussion of use of gas in the U. S. since its discovery in 1792 by Wm. Murdock an English Engineer. The first American use of gas illumination was in 1806 at Newport, R. I. Welsback mantle invented in 1880 by Dr. Welsback of Vienna jumped the candle power for a given flow of gas about 2000%. Electricity has now taken over many of the former uses of gas. "Coal reaching the heating and power plants in the form of gas is the ultimate fuel."

•

Pressure Losses for Fluid Flow in 90% Pipe Bends, by K. Hilding Beij. *Research Paper R. P.* 1110 *U. S. Dept. of Commerce, National Bureau of Standards, Journal of Research*, vol. 21, July 1938, page 1-18. Bibliography. This study forms part of an extended study of pipe bends in progress at the National Hydraulic Laboratory.

•

Comfort of Surgeons and Operating Room Personnel and/or Safety of Patients, by James Govan. *The Canadian Hospital*, vol. 15, no. 4, April 1938, page 16-19. A question and answer discussion of the comfort of surgeons and operating room personnel and possible methods of improving these conditions.

•

Air-Borne Infections, by William Firth Wells. *The Modern Hospital*, vol. 51, no. 1, July 1938. Discussion of the newer knowledge of air-borne infection. During coughing and sneezing, minute droplets containing micro-organisms from infected surfaces may be ejected into the air. Many of these evaporate before they can settle to the ground, leaving minute residues suspended in the air. These nuclei, in which the micro-organisms remain viable for considerable periods, may drift in air currents like particles of cigaret smoke. Methods of measuring sanitary ventilation. Bactericidal power of ultra-violet light screens.

•

Skin Temperature in Relation to the Warmth of the Environment, by Dr. T. Bedford. *Journal of Hygiene*, vol. 35, 1935, page 307-317. The skin temperatures of industrial workers, mainly women and girls, engaged in very light occupations, have been correlated with environmental conditions. Forehead, hand and foot temperatures were recorded.

ANNUAL REPORTS OF LOCAL CHAPTERS

Texas Chapter Grows Fast

Little more than a year ago when its charter was presented by Col. D. S. Boyden, the Texas Chapter had 42 members, and at the completion of its 1937-1938 season in May, 97 members were on the rolls:

The Officers elected to serve for 1938-1939 are as follows:

President—H. W. Skinner
Vice-President—C. L. Kribs, Jr.
Secretary—W. H. Badgett
Treasurer—R. K. Werner
Board of Governors—H. J. Martyn, C. A. McKinney and A. J. Rummel

The report of the Secretary, W. H. Badgett, points out that the Texas Chapter is unusual in that its members are scattered over a broad expanse of the largest state in the union, and during the past year in order to reach as many members as possible and stimulate interest in the Society activities, 10 dinner meetings were held in five cities—Dallas, Fort Worth, College Station, Houston, and San Antonio. The average attendance for the year was 38.

The largest meeting of the year was held March 5 on the occasion of President E. Holt Gurney's visit to Texas, and 71 members and guests attended the dinner at the Rice Hotel, Houston.

The splendid record of the Chapter was made under the guidance of the following Officers for 1937-1938:

President—R. F. Taylor
Vice-President—H. W. Skinner
Secretary—W. H. Badgett
Treasurer—I. E. Rowe
Board of Governors—M. L. Diver, C. L. Kribs, Jr., and J. A. Kiesling

October 9, 1937 in Dallas, E. K. Campbell, Kansas City, Mo., spoke on Warm Air Heating.

November 20, 1937 in Ft. Worth, H. W. Skinner, consulting engineer, Ft. Worth, talked about, A Yardstick for Weather.

January 15, 1938 in San Antonio, W. D. Masterson, Mgr., San Antonio Water Board, San Antonio, discussed The Relation of Air Conditioning in Texas to the State's Water Supply.

February 19, 1938 in Dallas, C. L. Kribs, Jr., consulting engineer, Dallas, was the speaker on, "Chiselling in Contracting," and movies of a Steam Jet Installation were shown.

March 5, 1938 in Houston, E. Holt Gurney, President, A.S.H.V.E., Toronto, gave an inspiring address on Air Conditioning Brings Human Problems to the Engineer.

March 22, 1938 in Ft. Worth, J. H. Van Alsburg, Chicago, Ill., talked on Comfort and Research.

April 2, 1938 in Dallas, Herman Seid, attorney for Auditorium Conditioning Corp., New York, spoke on Dollars and Sense of Air Conditioning.

April 11, 1938 in San Antonio, F. C. Houghten, director, A.S.H.V.E. Laboratory, Pittsburgh, gave an interesting discussion on The Society's Comfort Research Program.

April 12, 1938 in Houston, F. C. Houghten repeated his address on The Society's Comfort Research Program.

May 21, 1938 in College Station, F. E. Giesecke, 2nd Vice-President, A.S.H.V.E., College Station, was the featured speaker and his subject was The Influence of Building Construction on Air Conditioning.

Montreal Has Successful Year

The year's activities for the Montreal Chapter were recorded by R. R. Noyes, and 75 members are on the rolls.

For the 1938-1939 season, the following Officers will serve:

President—F. J. Friedman
Vice-President—L. H. Laffoley
Secretary—C. W. Johnson
Treasurer—F. G. Phipps
Board of Governors—George Ballantyne, A. B. Darling, Leo Garneau, W. U. Hughes, and G. L. Wiggs

A summary of the meetings held in 1937-1938 follows:

October 17, 1937 Windsor Hotel, attendance 42. Speaker—C. W. Walton, Jr., chief mechanical engineer of the operating office of the R. C. A. Building, Rockefeller Center, New York, gave a description of the Rockefeller Center Heating and Air Conditioning System.

November 15, 1937 Windsor Hotel, attendance 45. Speaker—Col. D. S. Boyden, President of the A.S.H.V.E., gave an illustrated lecture on The Economic Use of Purchased Steam.

December 13, 1937 Mount Royal Hotel, attendance 50. Speaker—G. L. Wiggs, consulting engineer and President of the local Chapter gave a detailed description of the Air Conditioning Systems in the Normandy Roof and Piccadilly Club of the Mount Royal Hotel.

January 17, 1938 Windsor Hotel, attendance 50. Speaker—F. A. Combe, consulting engineer, gave an illustrated talk on Fuels, Combustion and Boilers.

February 21, 1938 Windsor Hotel, attendance 44. Speaker—David L. Fiske, secretary of the *A.S.R.E.* gave a talk, the subject of which was Progress in Refrigeration.

March 21, 1938 Windsor Hotel, attendance 49. Speaker—Prof. E. A. Allcut, of the University of Toronto, gave an illustrated lecture on The Properties of Heat Insulating Materials for Buildings.

April 26, 1938 Windsor Hotel, attendance 23. Speaker—John Riley, of the accoustical department of Canadian Johns-Manville Limited, gave a talk on Noise and Sound Control.

May 16, 1938 Windsor Hotel, attendance 49. General business meeting with election of officers.

June 9, 1938 A golf meet and dinner was held at the Seigniory Club, Lucerne, Quebec; attendance 71.

The activities of the Chapter were directed during the most successful 1937-1938 season by the following Officers and Committees:

President—G. L. Wiggs
Vice-President—F. J. Friedman
Secretary—C. W. Johnson
Treasurer—F. G. Phipps
Board of Governors—A. B. Darling, Leo Garneau, W. U. Hughes, L. H. Laffoley and J. Linton
Nominating Committee—M. F. Dixon, F. A. Hamlet and T. Worthington
Golf Committee—John Colford, Chairman, A. B. Darling, Arthur Lamontagne and Tom Worthington

Pacific Northwest Chapter Meetings

The Officers of the Pacific Northwest Chapter are as follows:

President—C. W. May
Vice-President—R. O. Wesley
Secretary—R. D. Morse
Treasurer—E. E. Foote
Board of Governors—Lincoln Bouillon, W. W. Cox and S. D. Peterson

Chapter meetings were held each month and the following subjects were discussed:

January 7, 1938 Alaska Silver Millions by American Can Company.

February 4, 1938 Review of 44th Annual Meeting and high lights of 5th International Heating & Ventilating Exposition by C. W. May, delegate.

March 16, 1938 Air Conditioning by Parables, Lyle Dudley.

March 22, 1938 Air Conditioning Brings Human Problems to Engineers, E. Holt Gurney, President, A.S.H.V.E.

April 26, 1938 The Kaplin Turbines and Other Mechanical Features of Bonneville Project, Paul L. Heslop, Mechanical Engineer, U. S. Engineers.

May 10, 1938 Trip through U. S. Navy Yard, Bremerton, Wash.

June 14, 1938 Wrought Iron Production by Art Wicks.

A good attendance at all meetings and a record number on the membership rolls is reported by Secretary R. D. Morse.

Death of Prof. J. D. Hoffman

A host of friends from Coast to Coast mourn the loss of a stalwart figure in the field of heating and ventilating with the passing of Prof. J. D. Hoffman. His life was devoted to the advancement of the art of heating and ventilating and he taught thousands of engineers its fundamental principles and practical applications. In the many committees where he served his kindly advice and counsel will be sadly missed.

The Officers and Council of the Society are saddened by the death of a loyal and true friend and associate. Their heartfelt sympathy is extended to his family who survive.

James David Hoffman, past president and life member of the Society, died unexpectedly at his home in West Lafayette, Indiana, August 14.

On June 30, 1938, he had retired as professor of practical mechanics at Purdue University after completing 49 years as a teacher of engineering subjects. Professor Hoffman was a Hoosier and was born January 23, 1868, at Auburn, Ind. He attended public and high school in Auburn and then entered Purdue where he received the degree of B.M.E. in 1890 and the degree of M.E. in 1893. Professor Hoffman served an apprenticeship in the Auburn Foundry and Machine Works and was an expert in foundry work and pattern making, as well as in heating, ventilating and power plant equipment. He worked for the Buckeye Engine Co., Salem, O., and then joined the staff at Purdue in 1890. He served as instructor and assistant professor of practical mechanics, then machine design, and was associate professor of engineering design until 1911. He joined the College of Engineering at the University of Nebraska as professor of mechanical engineering and practical mechanics, and returned to Purdue as professor of mechanical engineering in 1918 and headed the department of practical mechanics until his retirement.

Professor Hoffman contributed many papers to technical societies and was the author of eleven books. He was a versatile writer and an able speaker.

In addition to the Handbook of Heating and Ventilation some of his contributions to A.S.H.V.E. TRANSACTIONS were: The Education of Heating and Ventilating Engineers; Notes on the Design of Central- Station Hot-Water Heating Systems; The Design of a Plenum System of Warm Air Heating for a School or Office Building; The Combined Pressure and Vacuum System of Steam Heating; Apparatus for the Study of Heat Radiation; Effect of the A.S.M.E. Boiler Code on Heating Boilers; and Reasons for Failures of Heating Systems.

He joined the A.S.H.V.E. in 1903, served as first vice-president in 1908, president in 1910, and on the Board of Governors during 1911 and 1912. He was elected to Life Membership in 1937. He was a life member of the *American Society of Mechanical Engineers*, the *Society for the Promotion of Engineering Education*, honorary member of the *National District Heating Association* and the *National Warm Air Heating and Conditioning Association*. He also belonged to Tau Beta Pi, Sigma Tau, and Sigma Xi.

Funeral services were held August 16 at Central Presbyterian Church with Dr. W. R. Graham officiating and interment was in Grand View Cemetery.

Air Conditioning Courses

The Polytechnic Institute of Brooklyn in New York City has revised its instructional work in the field of air conditioning which it has offered for several years, according to Prof. E. F. Church, Jr., Head of the Department of Mechanical Engineering. The schedule for the school year beginning in September offers the work in two courses. The first course, Air Conditioning, covers the fields of winter and summer conditioning in a practical manner and is open to students who have not had previous training of college grade and are desirous of training in the basic elements involved in the design, selection and operation of equipment.

The second course, Air Conditioning Engineering, is intended for those students who have had instruction in the fields of thermodynamics and other advanced technical fundamentals. This course covers the same field of study but deals comprehensively with theory and practice as employed therein.

Both courses extend throughout the year and include laboratory studies using the Institute's newly installed test equipment and its other laboratory facilities. This equipment employs many special features permitting the study of psychrometric effects over a wide range of conditions.

Promotion of W. A. Russell

The Hoffman Specialty Co. has announced the promotion of W. A. Russell to a position as General Sales Manager covering New England and New York State. Mr. Russell is a member of the Council and Chairman of the Speakers Bureau for the current year.